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SVIC NOTES

New Trends in Report Production

Research and development in any organization is documented in technical reports. In the past the various drafts of these reports passed through many hands on their way to becoming the printed word. In a typical scenario an author would write a manuscript in long hand, give it to the secretary who would type maybe one or two drafts of the report before sending it to the report production division for final editing, type setting and printing. Graphs and figures would be drawn in rough form by the author and in final form by a professional in the graphic services group. In this old mode there was ample opportunity for either secretary, editor or author to catch grammar and spelling errors.

Things are different today; most authors have access to personal computers (PC's) in their own office. Hardware and software are now available that allow the author to enter the text of a report directly into the computer; figures, graphs and tables are input directly into the computer using a "mouse," light pen or graphics tablet. And, with the introduction of the new laser printers, camera-ready material can be printed out with texts and figures interspersed on the author's own personal computer. Some have called these new capabilities a "home publishing" system; in some ways that is exactly what they are. Today a single author can sit down at the keyboard of a personal computer and quickly complete an entire publication, including figures and text, suitable for printing.

There are both advantages and disadvantages associated with these new report production methods. On the positive side there is obviously a great deal of time and money saved by having a single person create the entire camera-ready manuscript, test and figures, on a single computer. It also opens up the publishing business to anyone who can afford a five to ten thousand dollar investment. On the negative side, this will likely cause the elimination of some jobs in the centralized graphic services and report production sections of technical organizations. Also, there are many safeguards built into the old system where typists and editors went over each draft very carefully; errors were swept out before they got into print. Copyright infringement may become an issue with "home publishing" systems because the drafts are no longer reviewed by a professional editor. One of the functions of an editor is to seek permission to use previously copyrighted material. If the author borrows some material from a copyrighted work and forgets to ask permission, the author could be sued.

Research and development organizations must exercise a certain degree of caution when switching to these new modes of operation; some of the old safeguards, such as several stages of proofreading and editing by other than the author, should be retained. In order to keep up with the times, employees of research and development organizations will have to upgrade their skills in typing, spelling, grammar and computer-aided drafting.

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EDITORS RATTLE SPACE

EXPERT SYSTEMS FOR VIBRATION ANALYSIS

A term frequently encountered in today's world of high technology is expert system. As implied by the term it is a system whose purpose is to function as an expert. This capability has grown out of the advancing technology of digital computers, artificial intelligence, and pattern recognition. It is an interesting concept because it implies that computers can be trained to perform the logical processes that humans use in the design and analysis of complex engineering systems. In a more direct application, some researchers are attempting to apply all technology of expert systems to the analysis of vibration problems.

Engineers and vibration analysts do go through a logical process of reasoning when they try to solve a vibration problem. This involves the evaluation of facts about the machine, process, and environment along with data gathered from measurements. Much of the work is performed by the process of comparison elimination and/or pattern recognition is used to compare data acquired for the analysis to data previously related to known machine or structural condition. Through this method the vibration analyst eliminates or confirms the suspected source of the problem. One of the problems encountered by the vibration analyst is that many patterns obtained from FFT spectrum analyzers do not conform to known machine condition. To a large extent this occurs because of unknown natural frequencies and nonlinear system responses.

Computers and expert systems, in my opinion, will be limited to very basic vibration analysis until the vibration response of equipment is better understood by engineers. While progress is being made by engineers who develop diagnostics for specialized pieces of equipment such as electric motors, any person who has performed vibration analysis knows that new patterns show up each day from vibration measurements. It is important that potential purchasers of expert system software devise a number of check problems to measure the performance of such software against claims of capability. One must remember that the computers and black boxes do not perform magic and reasoning power beyond the capability of human beings is a myth. The expert system is only as good as its training. It will out perform the human being on the routine tasks of pattern recognition and complete logical checks to the extent of its data base.

R.L.E.

SYSTEM IDENTIFICATION USING REAL FREQUENCY-DEPENDENT MODAL CHARACTERISTICS

M. RADEŞ*

Abstract. Modal testing techniques in current use are based on the analysis of structural response in terms of either normal modes or complex modes of vibration. Methods based on characteristic phase lag modes or other frequency-dependent modal vectors are less used in experimental modal analysis and require considerable computation and knowledge of the system order. This paper treats methods suggested by Asher, Traill-Nash, Angélini, Fillod, Radeş, and Ibáñez based on frequency-dependent modal characteristics.

The dynamic response of systems with nonproportional damping can be expressed in terms of either complex modes or real characteristic phase-lag modes of vibration. The latter were introduced by Fraeijs de Veubecke [1] and apply only to harmonic forcing. The original approach and subsequent developments [2-6] are based on a generalized eigenvalue problem involving real and imaginary parts of the dynamic flexibility (DF) matrix.

A formulation by Traill-Nash [7] was extended by Radeş [8]; it utilizes the frequency response function (FRF) matrix, which is measurable and thus useful for identification purposes. Experimental [9] and numerical simulation [10] studies have been successful in determining undamped normal modes and modal parameters from identified characteristic phase-lag modes, at least for simple systems. The capability to identify structures with high damping and/or high modal density makes the technique attractive.

An equivalent approach, described in terms of the complex power transmitted to the tested structure, has been suggested by Angélini [11] and applied to the ground testing of aeronautical structures [12].

Another possibility is to use the characteristic eigenvectors of the real part of the FRF matrix [13]. The product of the corresponding eigenvalues equals the determinant used by Asher [14] to estimate undamped natural frequencies (UNF). His method was implemented [15] and extended [16] to rectangular FRF matrices. The singular value decomposition of these matrices has been used to establish system order, to locate UNF, and to determine tuned forcing vectors and truncated normal vectors [17,18].

The right singular vectors of the real part of the FRF matrix give the force distributions. According to the minimum coincident response method [19], these distributions minimize the singular values at the UNF.

For low order systems, an identification method has been derived [20] based on the eigenproblem of the real part of the DF matrix. Some difficulties are encountered in the accurate determination of this matrix from experimental data.

In all methods, plots of eigenvalues against frequency are used to accurately locate the UNF. Each characteristic eigenvalue vanishes only at the corresponding UNF. Thus, spurious resonance frequencies encountered with other methods can readily be eliminated. Modal parameter estimation is done using variants of the complex power method [21].

This paper presents the theoretical background of these methods. It also describes their merits and drawbacks and assesses their usefulness in modal testing.

FREQUENCY RESPONSE MATRICES

It is assumed that the motion of an actual structural system can be described by a linear constant parameter model having N degrees of freedom. The matrix equation is

$$[M]\,\{\ddot{q}\} + [C]\,\{\dot{q}\} + [K]\,\{q\} = \{f\} \tag{1}$$

System inertia, equivalent viscous damping, and stiffness matrices are denoted [M], [C] and [K] respectively. They are assumed real, symmetric, and positive definite.

For steady-state harmonic excitation $\{f\} = \mathrm{Re}\,(\{\check{f}\}\, e^{i\omega t})$ and response $\{q\} = \mathrm{Re}\,(\{\check{q}\}\, e^{i\omega t})$, the equation of motion becomes

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$$([K] - \omega^2 [M] + i\,\omega [C])\{\check{q}\} = \{\hat{f}\}$$

This can be written either

$$[B(\omega)]\ \{\check{q}\} = ([B^R(\omega)] + i[B^I(\omega)])\{\check{q}\} = \{\hat{f}\} \quad (2)$$

where $[B(\omega)]$ is the dynamic flexibility matrix, or

$$\{\check{q}\} = [H(\omega)]\{\hat{f}\} = ([H^R(\omega)] + i[H^I(\omega)])\ \{\hat{f}\} \quad (3)$$

$[H(\omega)]$ is the frequency response function matrix.

The element h_{ij} of the FRF matrix is the dynamic influence coefficient, defining the response at station i due to unit harmonic force at station j. The FRF matrix is measurable.

NORMAL MODES

Undamped normal modes $\{\Psi^{(r)}\}$ and corresponding undamped natural frequencies ω_r satisfy the eigenvalue equation

$$([K]-\omega_r^2[M])\{\Psi^{(r)}\} = [B^R(\omega_r)]\{\Psi^{(r)}\} = \{0\} \quad (r=1,2,\ldots,N) \quad (4)$$

Modal parameters of normal modes are defined by

$$m_r = \{\Psi^{(r)}\}^T[M]\{\Psi^{(r)}\}$$

$$k_r = \{\Psi^{(r)}\}^T[K]\{\Psi^{(r)}\} \quad (5)$$

$$c_{sr} = \{\Psi^{(s)}\}^T[C]\{\Psi^{(r)}\}$$

T denotes transpose.

The tuned forcing vector $\mathcal{F}^{(r)}$ gives the appropriated force distribution required to isolate the mode $\{\Psi^{(r)}\}$ at the frequency ω_r. This vector is given by

$$\{\mathcal{F}^{(r)}\} = \frac{1}{\gamma_{rr}}\omega_r[C]\{\Psi^{(r)}\} = \frac{1}{\gamma_{rr}}[B^I(\omega_r)]\{\Psi^{(r)}\} \quad (6)$$

where γ_{rr} is a scale factor.

CHARACTERISTIC PHASE LAG MODES

At any excitation frequency ω there exist $r=1,2,\ldots,N$ independent monophase force distribution $\{\hat{f}\}=\{F_\omega^{(r)}\}$. Each excites a corresponding monophase response mode $\{\Phi_\omega^{(r)}\}$ of forced vibration -- in which all points of the system vibrate in phase with one another -- at a characteristic phase lag $-\varphi_r(\omega)$ with respect to the excitation. As the frequency of excitation changes, so do the characteristic phase lags, response modes, and force distributions [1].

The equation that must be satisfied to meet the conditions for excitation of such a vibration is

$$\{\check{q}\} = \{\hat{q}\}e^{i\varphi} + [H^R + i\,H^I]\{\hat{f}\} \quad (7)$$

where $\{\hat{q}\}$ is real and φ is a characteristic phase angle. This is equivalent to considering that the out-of-phase portion $\{q^I\}$ of response is proportional to the in phase portion $\{q^R\}$; i.e., $\{q^I\}=\chi\ \{q^R\}$ where $\chi = \tan$ [6].

This leads to the following relations [7]:

$$([H^I] - \tan\ \varphi_r[H^R])\{F_\omega^{(r)}\}=\{0\} \quad (8) \quad (r= 1,2,\ldots,N)$$

$$([H^R]\cos\varphi_r+[H^I]\sin\varphi_r)\ \{F_\omega^{(r)}\} = \frac{1}{\gamma_r}\ \{\Phi^{(r)}\} \quad (9)$$

The first is an eigenvalue equation; the solutions yield the characteristic phase angles φ_r and associated monophase force vectors $\{F_\omega^{(r)}\}$. The monophase response vectors $\{\Phi^{(r)}\}$ are given by the second equation, in which $\gamma_r(\omega)$ is a scale factor.

At an undamped natural frequency ω_r the r-th response mode $\{\Phi^{(r)}\}$ becomes the normal mode $\{\Psi^{(r)}\}$; and its phase angle is $\varphi_r=-90^\circ$. The corresponding force distribution $\{F_{\omega_r}^{(r)}\}$ becomes the forcing vector $\{\mathcal{F}^{(r)}\}$ appropriated to the r-th normal mode and $\gamma_r(\omega_r)=\gamma_{rr}$.

The solutions $-\varphi_r(\omega)$, $\{\Phi_\omega^{(r)}\}$ and $\{F_\omega^{(r)}\}$ are termed the r-th characteristic phase lag mode [5]. Their frequency dependence is illustrated in Figure 1 for a two-degree-of-freedom system.

The force distribution modal vectors $\{\Gamma^{(r)}\} = \gamma_r(\omega)\{F_\omega^{(r)}\}$ satisfy the orthogonality relations

$$\{\Gamma_\omega^{(r)}\}^T[H^R(\omega)]\{\Gamma_\omega^{(s)}\}=0 \qquad (r\neq s)$$

$$\{\Gamma_\omega^{(r)}\}^T[H^I(\omega)]\{\Gamma_\omega^{(s)}\}=0 \qquad (10)$$

Two frequency-dependent modal characteristics are introduced

$$\alpha_r(\omega)=-\{\Gamma_\omega^{(r)}\}^T[H^I(\omega)]\{\Gamma_\omega^{(r)}\}=\{\Phi_\omega^{(r)}\}^T[B^I(\omega)]\{\Phi_\omega^{(r)}\}$$

$$(r=1,2,\ldots,N) \qquad (11)$$

$$\beta_r(\omega)=\{\Gamma_\omega^{(r)}\}^T[H^R(\omega)]\{\Gamma_\omega^{(r)}\}=\{\Phi_\omega^{(r)}\}^T[B^R(\omega)]\{\Phi_\omega^{(r)}\}$$

They are the real and imaginary parts respectively of the complex modal dynamic flexibility. Their frequency dependence is shown in Figure 2.

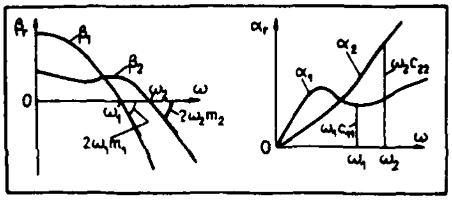


Figure 2. Frequency Dependence of Complex Modal Dynamic Flexibility.

During a cycle of vibration the complex energy transmitted to the structure by the excitation $\{\Gamma_\omega^{(r)}\}e^{i\omega t}$ is

$$W(\omega) = W_R(\omega)+iW_I(\omega) = i\pi\{\Gamma_\omega^{(r)}\}^T\{\Phi^{(r)}e^{i}{}_r(\omega) = i\pi\{\Gamma_\omega^{(r)}\}^T[H](\omega)]\{\Gamma_\omega^{(r)}\} \qquad (12)$$

it follows that

$$\sin\varphi_r = \frac{\alpha_r}{\sqrt{\alpha_r^2+\beta_r^2}} = \frac{W_R}{[W]},$$

$$\cos\varphi_r = \frac{\beta_r}{\sqrt{\alpha_r^2+\beta_r^2}} = \frac{W_I}{[W]} \qquad (13)$$

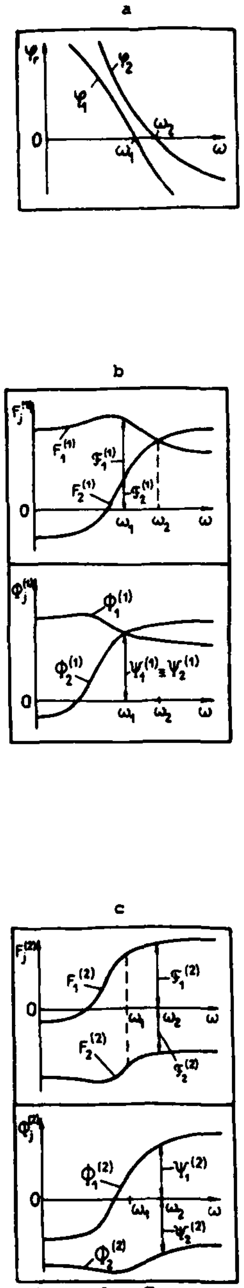


Figure 1. Frequency Dependence of R-th Characteristic Phase Lag Mode for a Two-Degree-of-Freedom System.

Thus $\sin \varphi_r$ is a measure of the relative active energy actually dissipated in the system. $\cos \varphi_r$ is a measure of the relative reactive energy. Their frequency dependence is shown in Figure 3.

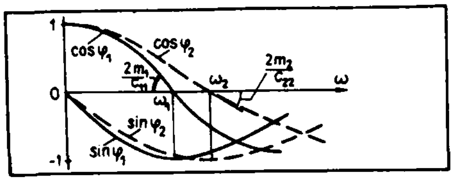


Figure 3. Frequency Dependence of Active and Reactive Energy in a System.

On the other side

$$\{\Gamma^{(r)}_{\omega}\}^T\{\Phi^{(r)}_{\omega}\} = \sqrt{\alpha_r^2=+\beta_r^2} \tag{14}$$

$$\{\Gamma^{(r)}_{\omega}\}^T\{\Phi^{(s)}_{\omega}\} = 0 \qquad (r\neq s)$$

Any monophase force vector develops energy only in the corresponding monophase response mode.

Equations (13) and (14) can be used to calculate $\alpha_r(\omega)$ and $\beta_r(\omega)$. the UNF ω_r are determined at the zero crossings of the diagrams $\beta_r(\omega)$ or $\cos \varphi_r(\omega)$.

Modal masses m_r of normal modes are proportional to the slopes at these points

$$m_r = -\frac{1}{\alpha\omega_r}\left(\frac{d\beta_r}{d\omega}\right)_{\omega_r} = \frac{c_{rr}}{2}\left(\frac{d\cos\ r}{d\omega}\right)_{\omega_r} \tag{15}$$

The diagonal modal damping coefficients c_{rr} are given by

$$c_{rr} = \frac{1}{\omega_r}\alpha_r(\omega_r) = \left(\frac{d\alpha_r}{d\omega}\right)_{\omega_r} \tag{16}$$

The appropriated force vectors $\{\mathcal{F}^{(r)}\}$ and normal mode vectors $\{\Psi^{(r)}\}$ are given by

$$[H^R(\omega_r)]\ \{\mathcal{F}^{(r)}\} = \{0\} \tag{17}$$

$$[H^I(\omega_r)]\ \{\mathcal{F}^{(r)}\} = -\frac{1}{\gamma_{rr}}\ \{\Psi^{(r)}\} \tag{18}$$

The quantity $[H(\omega_r)]$ is calculated either by interpolation, based on already recorded experimental data taken at other ω, or from a new series of measurements taken at ω_r.

The non-diagonal damping coefficients c_{rs} are obtained from

$$c_{rs} = \frac{1}{\omega_r}\gamma_{rr}\{\Psi^{(s)}\}^T\{\mathcal{F}^{(r)}\} \tag{19}$$

The procedure offers a means for determining normal modes from measured FRF data for systems with nonproportional damping. Monophase force vectors and response modes become independent of frequency in the case of proportional damping. This property is the basis for the only practical method for ascertaining the existence of proportional damping in a structure [10].

Characteristic phase lag modes are independent of the type of damping. The same for any combination of linear viscous and hysteretic damping, they are forced modes of vibration defined only for harmonic excitation; they cannot be used in a transient or shock response analysis.

EIGENVECTORS OF THE COINCIDENT FRF MATRIX

A simpler modal parameter extraction method [13] is based on the eigenvalue problem of the real part of the FRF matrix

$$[H^R(\omega)]\{x^{(r)}_{\omega}\} = \mu_r(\omega)\{x^{(r)}_{\omega}\} \qquad (r=1,2,\ldots,N) \tag{20}$$

If the system is acted upon by the monophase force distribution $\{x^{(r)}_{\omega}\}e^{i\omega t}$ the eigenvalue $\mu_r(\omega)$ is a measure of the reactive energy transmitted to the system. This energy cancels at the undamped natural frequency ω_r.

The forcing vectors $\{x^{(r)}_{\omega}\}$ satisfy the orthogonality conditions

$$\{x^{(r)}_{\omega}\}^T\{x^{(s)}_{\omega}\} = 0 \qquad (r\neq s)$$

$$\{x^{(r)}\}^T[H^R(\omega)]\{x^{(s)}_{\omega}\} = 0$$

Solving equation (20) at discrete values ω, for which $[H^R(\omega)]$ is measured, and plotting the eigenvalues μ_r against frequency (Figure 4), the UNF ω_r can be located at the zero crossings: $\mu_r(\omega_r) = 0$.

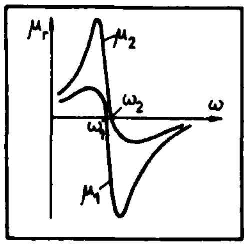


Figure 4. Location of Undamped Natural Frequencies

For $\omega = \omega_r$ equation (20) becomes identical with equation (17); $\{x_\omega^{(r)}\} \equiv \{\mathcal{F}^{(r)}\}$ is thus the tuned forcing vector that can isolate the normal mode $\{\psi^{(r)}\}$ at ω_r. The latter is obtained from equation (18) together with the scale factor γ_{rr}. Because

$$\det[H^R(\omega)] = \prod_{r=1}^{N} \mu_r(\omega),$$

it follows that

$$\det[H^R(\omega_r)] = 0.$$

This is the basis of Asher's method [14].

Denote

$$\overline{\beta}_r(\omega) = \{x_\omega^{(r)}\}^T[H^R(\omega)]\{x_\omega^{(r)}\} =$$

$$\mu_r\{x_\omega^{(r)}\}^T\{x_\omega^{(r)}\} \tag{21}$$

$$\overline{\alpha}_r(\omega) = -\{x_\omega^{(r)}\}^T[H^I(\omega)]\{x_\omega^{(r)}\}$$

to obtain $\overline{\beta}_r(\omega_r) = 0$

$$\left(\frac{d\overline{\beta}_r}{d\omega}\right)_{\omega_r} = -\frac{2\omega_r m_r}{\gamma_{rr}^2} =$$

$$\{\mathcal{F}^{(r)}\}^T\{\mathcal{F}^{(n)}\}\left(\frac{d\mu r}{d\omega}\right)_{\omega_r}$$

$$\alpha_r(\omega_r) = \frac{\omega_r c_{rr}}{\gamma_{rr}^2}$$

Values for m_r and c_{rr} can be obtained from this relationship.

The quadratic forms of equation (21) are proportional to the reactive and the active energy transmitted to the system.

If vectors $\{\mathcal{F}^{(r)}\}$ are orthonormalized, modal masses of normal modes given by

$$m_r = -\frac{\gamma_{rr}^2}{2\omega_r}\left(\frac{d\mu_r}{d\omega}\right)_{\omega_r} \tag{22}$$

are proportional to the slope of the $\mu_r(\omega)$ diagram at the point of frequency axis crossing. Non-diagonal modal damping coefficients are given by equation (19).

The characteristic eigenvalue plots permit a more accurate location of the UNF than the determinantal plot used in Asher's method. The technique can be used to separate vibration modes with closely spaced UNF and can identify modes with overcritical damping [22]. The main drawback is the requirement to determine, in a preliminary stage, the order of the system and to use square FRF matrices.

SINGULAR VECTORS OF THE COINCIDENT FRF MATRIX

When the number of response measurement points exceeds the number of excitation points, the real part of the FRF matrix, denoted $[H^R(\omega)]$, is rectangular. The following singular value problem can be set up

$$[H^R(\omega)]\{u_\omega^{(r)}\} = \rho_r(\omega)\{v_\omega^{(r)}\}$$

$$[H_*^R(\omega)]^T\{v_\omega^{(r)}\} = \rho_r(\omega)\{u_\omega^{(r)}\} \tag{23}$$

where $_r$ are singular values, $\{u^{(r)}\}$ are orthonormalized right singular vectors, and $\{v^{(r)}\}$ are orthonormalized left singular vectors.

Adequate algorithms and numerical procedures are available for the singular value decomposition of the FRF matrix. Singular values are fairly insensitive to perturbations in the matrix elements; the eigenvalues of certain experimentally constructed square matrices are very sensitive.

Equations (23) yield

$$\rho_r^2(\omega)=\{u_\omega^{(r)}\}^T[H^R(\omega)]^T[H_*^R(\omega)]\{u_\omega^{(r)}\} \tag{24}$$

This relationship resembles the error function used in the minimum coincident response method [19]; i.e., the sum of the squares of the coincident responses. If follows that the right singular vector $\{u_\omega^{(r)}\}$ evaluated at ω_r provides the force distribution that minimizes ρ_r.

Plots of the singular values against frequency exhibit minima at some or all of the UNF. Figure 5 is such a plot for a three-degree-of-freedom system. The system order equals the number of nonzero singular values.

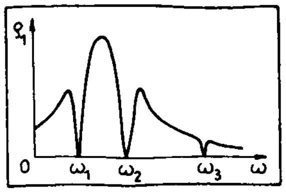


Figure 5. Minima of Undamped Natural Frequencies for a Three-Degree-of Freedom System.

It has been shown [18] that complete tuned forcing vectors can be derived from the pseudo-inverse of a submatrix of $[H_*^R(\omega)]$. In the case of incomplete excitation, truncated undamped modes of vibration can be analytically obtained using the imaginary part of the FRF matrix $[H_*^I(\omega)]$ and the appropriated forcing vectors. The structures need not actually be excited by them.

EIGENVECTORS OF THE COINCIDENT D.F. MATRIX

An alternative method can be similarly derived using the eigenvalue problem of the real part of the dynamic flexibility matrix

$$[B^R(\omega)]\{y_\omega^{(r)}\} = \lambda_r(\omega)\{y_\omega^{(r)}\} \qquad (r=1,2,\ldots,N) \tag{25}$$

If a complex force distribution produces a monophase response vector $\{y_\omega^{(r)}\}$ that is an eigenvector of the matrix $[B^R(\omega)]$, the corresponding eigenvalue $\lambda_r(\omega)$ is a measure of the reactive energy transmitted to the system by that excitation. The eigenvalue cancels at ω_r; that is, $\lambda_r(\omega_r)=0$. The response vectors $\{y_\omega^{(r)}\}$ satisfy the orthogonality conditions

$$\{y_\omega^{(r)}\}^T\{y_\omega^{(s)}\} = 0 \qquad (r \neq s)$$
$$\{y_\omega^{(r)}\}^T[B^R(\omega)]\{y_\omega^{(s)}\} = 0$$

Equations (4) and (25) show that $\{y_\omega^{(r)}\}$ is the eigenvector that becomes $\{\psi^{(r)}\}$ at ω_r [20]. Again

$$\det[B^R(\omega)] = \prod_{r=1}^{N} \lambda_r(\omega) \qquad \text{and}$$
$$\det[B^R(\omega_r)] = 0$$

The quadratic forms

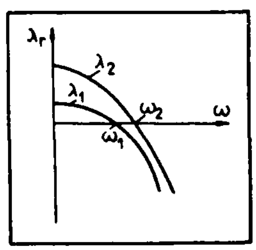
$$\beta_r(\omega)=\{y_\omega^{(r)}\}^T[B^R(\omega)]\{y_\omega^{(r)}\} = \lambda_r\{y_\omega^{(r)}\}^T\{y_\omega^{(r)}\}$$

$$\alpha_r(\omega)=\{y_\omega^{(r)}\}^T[B^I(\omega)]\{y_\omega^{(r)}\} \tag{26}$$

are introduced to obtain $\beta_r(\omega_r) = 0$

$$\left(\frac{d\beta r}{d\omega}\right)_{\omega_r} = -2\,\omega_r m_r = \{\psi^{(r)}\}^T\{\psi^{(r)}\}\left(\frac{d\lambda r}{d\omega}\right)_{\omega_r}$$

$$\alpha_r(\omega_r) = \omega_r c_{rr}$$



Figrue 6. Plots of $\lambda_r(\omega)$ vs Frequency

Plots of $\lambda_r(\omega)$ against frequency (Figure 6) exhibit zeros at the UNF ω_r. Modal masses of normal modes are given by

$$m_r = -\frac{\{\psi^{(r)}\}^T\{\psi^{(r)}\}}{2\,\omega_r}\left(\frac{d\lambda r}{d\,\omega}\right)_{\omega_r} \tag{27}$$

Modal damping coefficients are given by

$$c_{rs} = \frac{1}{\omega_r} \{\Psi^{(s)}\}^T [B^I(\omega)_r] \{\Psi^{(r)}\} \tag{28}$$

Matrices $[B^R(\omega)]$ and $[B^I(\omega)]$ can be obtained from measured FRF data using the conjunctive transformations

$$[B^R(\omega)] = [H^*(\omega)]^{-1} [H^R(\omega)][H(\omega)]^{-1}$$

$$[B^I(\omega)] = -[H^*(\omega)]^{-1} [H^I(\omega)][H(\omega)]^{-1}$$

The asterisk denotes the conjugate transpose. For large systems the inversion can lead to ill conditioning.

CONCLUSIONS

The methods presented permit the determination of normal modes from identified frequency-dependent modal vectors for systems with general nonproportional damping. They have the capability of separating vibration modes with closely spaced undamped natural frequencies and/or overcritical damping.

All methods are based on the solution of an eigenvalue problem at a series of excitation frequencies. Solution involves storing and manipulating a large volume of data; substantial peripheral storage is required. Precise location of UNF and evaluation of the FRF matrix at these frequencies are critical for the accuracy by which modal parameters are estimated. As has been mentioned [23], good results can be obtained using analytically synthesized FRF's, including residual terms, rather than experimentally-obtained FRFs.

All modal characteristics are calculated from data obtained by non-appropriated harmonic excitaion. Use of the complete FRF or DF matrices permits determination of global estimates of modal parameters. Consistent FRF data have to be obtained by multi-point excitation and parallel multichannel data collection.

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LITERATURE REVIEW: survey and analysis of the Shock and Vibration literature

The monthly Literature Review, a subjective critique and summary of the literature, consists of two to four reviews each month, 3,000 to 4,000 words in length. The purpose of this section is to present a "digest" of literature over a period of three years. Planned by the Technical Editor, this section provides the **DIGEST** reader with up-to-date insights into current technology in more than 150 topic areas. Review articles include technical information from articles, reports, and unpublished proceedings. Each article also contains a minor tutorial of the technical area under discussion, a survey and evaluation of the new literature, and recommendations. Review articles are written by experts in the shock and vibration field.

HELICOPTER VIBRATION CONTROL — RECENT ADVANCES

G.T.S. Done*

Abstract. **This article describes recent advances in the control of helicopter vibration and reviews associated literature. Vibration absorbers and isolators, direct rotor control, structural design and modification, and vibration studies are considered.**

Earlier articles [1,2,3] reviewed progress in and literature on helicopter vibration control up to mid 1982. The present paper covers the last three years. It is concerned only with airframe forced vibration and does not include consideration of aeroelastic instability phenomena or noise.

Although papers that provide descriptions of the vibratory airloading of helicopter rotors (the source of the vibrations) fall outside the scope of this review article, mention should be made of the excellent work of Hooper [4], who has provided a valuable contribution in this respect. The scene in general has been surveyed by Loewy [5] in an article covering all aspects of helicopter vibration including control from the 1950s to the present. Another review paper [6] tends to concentrate on control by structural modification and direct rotor control.

VIBRATION ABSORBERS AND ISOLATORS

The main source of vibration on a helicopter is the rotor. The conventional position for vibration isolators is between the fuselage and the main gearbox. Soft isolation is unacceptable because large relative deflections that arise from various loading situations between gearbox and rotor and gearbox and the fuselage create engineering difficulties. Devices that provide a stiff mounting have therefore been developed; they rely on converting the small relative displacement between gearbox and fuselage into a large opposite displacement of a small mass. The result is a momentum balance of the moving components at the tuning frequency. Isolation is in effect provided by force cancellation. One of the most well known devices is the DAVI (Dynamic Antiresonant Vibration Isolator), which has previously been described [3].

An account of ground and flight tests of an isolation system based on four such devices (the SARIB vibration absorber) has been given by Hege and Genoux [7]. This paper also provides a useful table of all vibration control devices and systems that have been used by major helicopter manufacturers since 1965. Similar tests on a similar isolation system (ARIS -- Anti-Resonance Isolation System) have also been reported by Braun [8]. He measured significant vibration reductions; the added isolation system weight was only one percent of the maximum takeoff weight. Two types of isolator were tested. One had the balance mass on a mechanical pendulum; on the other (Figure 1) the mass was driven hydraulically. The latter is used in a lateral sense when fitted to a helicopter.

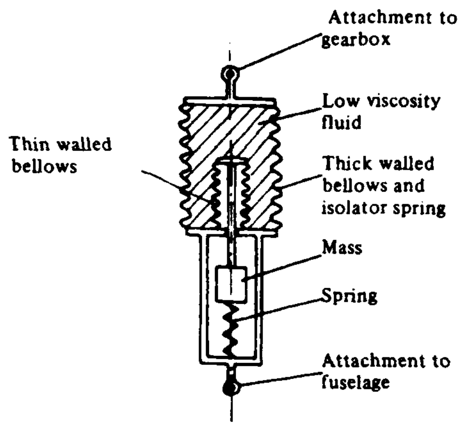

Figure 1. Antiresonance Force Isolator with Hydraulically Driven Balance Mass

The application and testing of a vibration absorber similar to a mechanical bi-filar pendulum absorber have been described by Viswanathan and McClure [9]. The pendulum mass is replaced by a quantity of mercury. In effect, the cylindrical container wall forms the pendulum arm.

Two absorbers can be embodied in one unit (Figure 2); in this case the absorber is designed to control the 4P (four times rotor speed) and 3P vertical shear vibrations. Cabin vibration has been reduced by 60 percent in flight tests. A

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persistent problem of rotor-borne absorbers is reliability. King [10] has described the way in which the particular design of an absorber -- in effect, a sprung mass -- has considerably reduced overall defects reported in service.

Absorbers can also be mounted in the fuselage. The best locations for three damped spring-mass absorbers on the cabin floor of a 39 degrees-of-freedom helicopter fuselage model have been described [11]. Stoppel [12] has critically analyzed calculated and test results for the ARIS system [8].

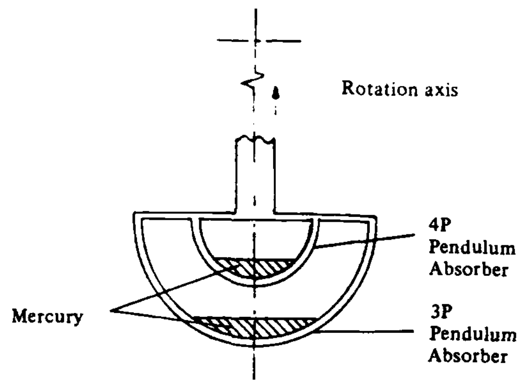

Figure 2. Mercury Pendulum Absorber

DIRECT ROTOR CONTROL

In direct rotor control, signals are fed to actuators that provide additional pitch deflections on the rotor blades. The actuators may be above or below the swash plate. This is an active control system; deflections are generally at once per blade passing frequency and other blade frequency multiples -- hence the term higher harmonic control, or HHC. This method of active control is not only useful for reducing vibration in flight (via pitch-flap coupling) but is also used for gust alleviation and control of retreating blade stall. Only references that provide information on the vibration aspect are quoted here.

The method has seen considerable development over the past decade. Results of flight tests have shown significant improvements in vibration performance [13,14]. Comprehensive wind-tunnel tests carried out on fully representative models [15-17] also demonstrate improvements. One paper [18] provides further information on another [15]. Molusis [19] has sought to explain why HHC does not reduce certain types of vibration that have been reported [18] as much as had been expected.

Ham [20] has published the theoretical background for a system of individual blade control (IBC); vibration is mentioned briefly. Results of wind-tunnel tests have not yet been reported. The problem of embodying fuselage modes into a theoretical model when HHC is used has been addressed [21]. Existing active vibration controllers have been evaluated [22]; suggestions for improvements were also given.

STRUCTURAL DESIGN AND MODIFICATION

The passive control of vibration by structural design or modification has been discussed in a survey paper [23] that categorizes and compares the various approaches available as either vibration reduction in the rotor or vibration reduction in the fuselage. Formal optimization techniques have been used in the former and to some extent in the latter. A review [24] of applications of optimization in helicopter design covers the vibration control problem.

Friedmann and Shanthakumaran [25,26] used optimization in a blade design; a reduction of 15-40 percent of the vibratory hub shears in forward flight could be achieved compared with the baseline design. In addition, there was a reduction of 9-20 percent in blade weight. Davis [27] extended work based on modal shaping [28] by coupling it to a formal optimization procedure. Bennett [29] devoted part of a paper to the problem of reducing vertical shear vibration at the rotor head. Peters et al [30] concentrated on frequency placement; they assumed that vibration can be reduced by separation of blade flap, lag, and torsional rotating natural frequencies from various frequency components of aerodynamic excitation that occur at multiples of rotor speed. Fuselage vibration has been linked directly with rotor-system design parameters in a preliminary optimization study [31].

Other examples of blade tailoring do not use formal optimization techniques or change normally available blade design parameters. Gorman and Sharpe [32] have explored the feasibility of introducing local compressive prestressing to reduce vertical shear at the rotor head. Kottapalli [31] has shown that the applied angular deflection of a fixed blade tab can provide a significant reduction. The effect of blade sweep on rotor vibratory loads has been shown [34]. Variable success for a tuned (i.e. flexible) tab attached to the outer part of a rotor blade has been reported [35].

In the area of fuselage modification, dynamic equations have been formulated such that anti-resonance can be satisfied [36]; these equations were applied to a simple helicopter fuselage model using added beams for absorbers. Dowell [37] has commented on and added to an earlier approach [38] for determining changes in modal characteristics due to structural modification. Sobey [39] has extended the Vincent circle technique to deal with multiple force inputs and a required minimum weighted mean square response at a number of locations throughout an airframe over a range of forward speed.

A paper by Niebanck [40] allows much of the work reported in this section to be seen in perspective. He presents experimental results from six sources that show the effect of blade mass tuning or other structural modification on rotor head loads and fuselage vibration. Many analytical predictions are not borne out in practice.

VIBRATION STUDIES

Alleviation of vibration in the fuselage requires a sound understanding of the vibration properties of the fuselage and of the combined fuselage/rotor system. A contribution to the latter includes in the analysis in-plane degrees of freedom [41]. The degree of complexity currently necessary to model a helicopter fuselage has been described [42]; the accuracy with which forced vibration response can be predicted was also shown. A procedure for combining experimentally measured dynamics of wing mounted stores and analytical fuselage dynamics has been described [43]. It has been shown that a physical model of a fuselage can be designed so that it correctly scales dynamically a much more complex model [44]; optimization and identification techniques were used.

CONCLUDING REMARKS

It is interesting to examine the variation in the number of papers in each category since 1974 [1-3] and the present; the variation reflects in a broad way the research and development activities under the various headings. For example, activity on mechanical devices for vibration control, such as isolators and absorbers, would seem to be on the decrease. This is counterbalanced by an ever-increasing amount of work in structural design and modification, in particular in the area of rotor-blade tailoring. These changes should not be unexpected because use of optimization techniques in engineering is now widespread. In addition, directing attention to blades rather than use devices is reasonable because the problem is being attacked closer to its source. Direct rotor control, which also allows problems to be tackled at source, has shown a steady rate of activity with continual progress over the past eight or nine years. Vibration studies can be regarded as being in support of other activities and, apart from a spate of activity three to six years ago, continue at a fairly low level.

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BOOK REVIEWS

SEISMIC EVENTS PROBABILISTIC RISK ASSESSMENTS

P.Y. Chen and C.I. Grimes, eds.
ASME Pub., PVP - Vol 79
American Society of Mechanical Engineers
New York, NY
1984, 75 pp

Until recently, quantitative probabilistic risk assessment (PRA) techniques have been used only in evaluations of commercial nuclear power plants. Seismic risk analysis usually includes seismic hazard analysis, fragility determination of structures, systems and accident sequence analysis, and consequence analysis. As stated by the editors, "The purpose of this volume is to present, as concisely as possible, the various state of the art of seismic risk analysis, lessons learned so far and valuable recommendations for future improvements."

The symposium consisted of nine papers. The initial paper contains an assessment of seismic-induced pipe break probability in PWR reactor coolant loops. Federal requirements are that nuclear power plants be designed for the effects of an unlikely event of a double-ended guillotine break (DEGB) of reactant coolant loop (RCL) piping. The conclusions were that the probability of indirectly induced DEGB in RCL piping due to earthquake is very small for Westinghouse reactors and that, if a piping system rupture occurs, the most likely location is inside the reactor ca ity.

The second paper has to do with the failure probability of PWR reactor/coolant loop piping. For direct DEGB consideration is given to the probability of the existence of cracks, crack size distribution, and hydrostatic proof tests. The next paper considers PRA in structural fragility and seismic probabilistic risk assessment for BWR/6 Mark III in a standard plant and contains an evaluation of structural fragility of safety-related structures in a Mark III standard plant.

The fourth paper presents a reliability analysis method for safety evaluation of nuclear structures. Ground acceleration due to an earthquake is represented by a segment of a stationary Gaussian process with zero mean and a Kanai-Tajimi spectrum. Results of studies of a containment subjected to dead load, live load, and ground accelerations from an earthquake are presented in the form of a fragility curve for PRA studies. The next paper identifies the sensitive nuclear power plant components using PRA. The sixth paper reviews seismic PRA. Seismic events affect the frequency of core melt and other risks. Areas of concern are seismic, hazard/structure fragility curve integration, generic plant specific fragility data, and errors in design and construction.

The next paper describes, characterizes, and evaluates sources of uncertainty for seismic risk. The current state of the art does not permit the following variables in probabilistic safety analysis procedures: plant familiarization, accident sequence definition, reliability data assessment, accident sequence quantification, hazard analysis, fragility analysis, system analysis, and quantification and interpretation of results. Advances must combine uncertainty, sensitivity, and importance. The eighth paper assesses probabilistic risks and considers weak and strong points of PRA.

The last paper describes a broad seismic hazard analysis model that is simple compared to the present use of PRA. All of the parameters are bound by uncertainty. The focus is on reducing indeterminacies. The author considers this uncertainty and variability in seismic hazard analysis.

The uncertainties and variabilities in PRA must be clarified. Researchers and investigators must strive to make PRA a creditable subject. The reviewer recommends this volume to those interested in the probabilistic risk assessment of seismic hazards.

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GUIDE FOR INDUSTRIAL NOISE CONTROL

P.N. Cheremisinoff and F. Ellerbusch
Butterworth Publishers, Ann Arbor Science
Ann Arbor, MI
1982, 190 pp, $29.95

The title of this book reminded me of the last book I read by Mr. Cheremisinoff, Industrial Noise Control Handbook, also published by Ann Arbor Science. It contains almost 400 pages and is hard cover. I found it inadequate as a handbook because of the sketchy treatment of the few topics discussed. The book under review, Guide for Industrial Noise Control, has nine chapters; six of them were not written by those named on the cover, and only two are concerned with details of noise control.

An introductory chapter on sound and noise is too brief and states as fact some controversial issues: that hearing is less important than eyesight, the speech frequencies range from 20-2,000 Hz, and that vibrations in a medium result in sound. An introductory chapter should not state that combining two equal levels results in a 3dB increase without any explanation.

The definitions are confusing. Power is defined as energy per unit time and intensity as energy per unit area. The equation, Power = Intensity x Area, is correct because, in a simplistic sense, intensity is equal to power per unit area. The smart reader would stop reading at this point; unfortunately, the reviewer must continue.

The second chapter is an overview of industrial noise control. Important points include the valuable, but often understated, concept that the best noise control is achieved with proper maintenance and that noise control starts with the purchase of quiet equipment. The person with an immediate noise control problem would not find help in this chapter, however. The half page treatment of transmission loss cannot help but confuse a reader new to the field.

The next chapter describes noise control regulations. Noise control when this review was written in the winter of 1986 was not emphasized in state, local, and federal regulations. Nevertheless, when the book went to press, concern was being expressed; the short treatment given to EPA, for example, might have been of interest then but is not now. The chapter also includes agency government research. In short the chapter can be skipped. I suggest the potential reader borrow or buy a good noise control handbook, get a copy of current state or federal regulations, or ask knowledgeable people for guidance.

The fourth chapter on noise analysis was written by an author not named on the cover, as are the rest of the chapters. The chapter is interesting and useful as an introduction to measurements and instrumentation. The fifth chapter deals with vibration analysis and instrumentation. It presents a simple and somewhat dated picture of the vibration measurement area.

In the sixth chapter on measurement techniques for sound level meters the author implies that Type 2 meters are adequate for most situations but does not define the situations. It is my opinion that noise measurements must be done in a standard way if results of two or more measurements are to be comparable. ASTM, ANSI, and ISO standards are often the techniques to use. I believe the author is remiss in not discussing these techniques or at least providing appropriate citations to the standards.

The seventh chapter deals with audiometry. The eighth chapter on valve/piping noise contains a reasonable treatment that can be supplemented by other books. The last chapter is on fan noise; it is short and not sufficiently detailed to use as a stand-alone section. A glossary and index end the book.

So what of enclosure design, gear noise, damping, reverberation control, and ear protectors. To learn of them search elsewhere -- they won't be found in this Guide.

My conclusions are evident. You can't judge a book by its title. If you have a choice, buy one or more of the other books available. If the books are not available in a bookstore or from the publisher, call a consultant or a knowledgeable person in the field and ask for assistance.

R.J. Peppin
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SHORT COURSES

SEPTEMBER

TIME DOMAIN MODAL VIBRATION TESTING TECHNIQUE
Dates: September 8-9, 1986
Place: Virginia Beach, Virginia
Objective: The seminar presents an in-depth study of the ITD method, examining results of previous applications and the applied use of the computer program and its selected options. Through attendance at the workshop, participants will receive the complete computer program of the ITD method and should be able to use the technique in modal vibration testing applications.

Contact: W.C. Bentley, Industrial Programs, School of Engineering, Old Dominion University, Norfolk, VA 23508 - (804) 440-4243

VIBRATION DAMPING TECHNOLOGY
Dates: September 15-19, 1986
Place: Dayton, Ohio
Dates: January, 1987
Place: Clearwater, Florida
Objective: Basics of theory and application of viscoelastic and other damping techniques for vibration control. The courses will concentrate on behavior of damping materials and their effect on response of damped systems, linear and nonlinear, and emphasize learning through small group exercises. Attendance will be strictly limited to ensure individual attention.

Contact: David I. Jones, Damping Technology Information Services, Box 565, Centerville Branch USPO, Dayton, OH 45459-9998 - (513) 434-6893.

PENETRATION MECHANICS
Dates: September 15-19, 1986
Place: Southwest Research Institute
San Antonio, TX
Objective: This course covers the full spectrum of problems encountered in penetration mechanics. It has been structured to emphasize the physical basis for analyzing and solving problems in penetration dynamics from low velocities to hyper velocities. The solution techniques will be applied to real problems to obtain insight and understanding to impact phenomena. The tentative outline below describes the course coverage in more detail.

GEAR NOISE
Dates: September 17-19, 1986
Place: The Ohio State University
Objective: The course will cover general noise measurements and analysis, causes of gear noise, gear noise reduction techniques, dynamic modeling, gear noise signal analysis, and modal analysis of gear boxes. Problems of course attendees will be discussed in special workshop sessions. Laboratory demonstrations will also be given. Featured speakers will be Mr. Donald Welbourn formerly of The University of Cambridge, and Professors D.R. Houser and R. Singh of The Gear Dynamics and Gear Noise Research Laboratory at The Ohio State University.

Contact: Mr. Richard D. Frasher, Director, Continuing Education, College of Engineering, 2070 Neil Avenue, Columbus, Ohio 43210, (614) 422-8143

OCTOBER

RANDOM VIBRATION IN PERSPECTIVE — AN INTRODUCTION TO RANDOM VIBRATION AND SHOCK, TESTING, MEASUREMENT, ANALYSIS, AND CALIBRATION, WITH EMPHASIS ON STRESS SCREENING
Dates: October 6-10, 1986
Place: Boston, Massachusetts
Dates: November 3-7, 1986
Place: Orlando, Florida
Dates: February 2-6, 1987
Place: Santa Barbara, CA
Dates: March 9-13, 1987
Place: Washington, D.C.
Dates: April 6-19, 1987
Place: Ottawa, Ontario
Objective: To show the superiority (for most applications) of random over the older sine vibration testing. Topics include resonance, accelerometer selection, fragility, shaker types,

fixture design and fabrication, acceleration/power spectral density measurement, analog vs digital controls, environmental stress screening (ESS) of electronics production, acoustic (intense noise) testing, shock measurement and testing. This course will concentrate on equipment and techniques, rather than on mathematics and theory. The 1984 text, "Random Vibration in Perspective," by Tustin and Mercado, will be used.

Contact: Wayne Tustin, 22 East Los Olivos St., Santa Barbara, CA 93105 - (805) 682-7171.

1986 JOHN C. SNOWDON VIBRATION CONTROL SHORT COURSE
Dates: October 20-24, 1986
Place: The Pennsylvania State University
Objective: This course, under the sponsorship of the Applied Research Laboratory, is presented by internationally known lecturers. It was initiated by the late Professor John C. Snowdon a decade ago and now continues under the guidance of Dr. Eric E. Ungar of Bolt Baranek and Newman, Inc. The course emphasizes principles, general approaches and new developments, with the aim of providing participants with efficient tools for dealing with their own practical vibration problems.

Contact: Gretchen A. Leathers, 410 Keller Conference Center, University Park, PA 16802 - (814) 863-4563

NOVEMBER

MACHINERY VIBRATION ANALYSIS I
Dates: November 11-14, 1986
Place: Chicago, Illinois
Objective: This course emphasizes the role of vibrations in mechanical equipment instrumentation for vibration measurement, techniques for vibration analysis and control, and vibration correction and criteria. Examples and case histories from actual vibration problems in the petroleum, process, chemical, power, paper, and pharmaceutical industries are used to illustrate techniques. Participants have the opportunity to become familiar with these techniques during the workshops. Lecture topics include: spectrum, time domain, modal, and orbital analysis; determination of natural frequency, resonance, and critical speed; vibration analysis of specific mechanical components, equipment, and equipment trains; identification of machine forces and frequencies; basic rotor dynamics including fluid-film bearing characteristics, instabilities, and response to mass unbalance; vibration correction including balancing; vibration control including isolation and damping of installed equipment; selection and use of instrumentation; equipment evaluation techniques; shop testing; and plant predictive and preventive maintenance. This course will be of interest to plant engineers and technicians who must identify and correct faults in machinery.

Contact: Dr. Ronald L. Eshleman, Director, The Vibration Institute, 101 West 55th Street, Suite 206, Clarendon Hills, IL 60514 - (312) 654-2254.

ABSTRACTS FROM THE CURRENT LITERATURE

ABSTRACT CONTENTS

AVAILABILITY OF PUBLICATIONS ABSTRACTED

None of the publications are available at SVIC or at the Vibration Institute, except those generated by either organization.

Periodical articles, society papers, and papers presented at conferences may be obtained at the Engineering Societies Library, 345 East 47th Street, New York, NY 10017; or Library of Congress, Washington, D.C., when not available in local or company libraries.

Government reports may be purchased from National Technical Information Service, Springfield, VA 22161. They are identified at the end of bibliographic citation by an NTIS order number with prefixes such as AD, N, NTIS, PB, DE, NUREG, DOE, and ERATL.

Ph.D. dissertations are identified by a DA order number and are available from University Microfilms International, Dissertation Copies, P.O. Box 1764, Ann Arbor, MI 48108.

U.S. patents and patent applications may be ordered by patent or patent application number from Commissioner of Patents, Washington, D.C. 20231.

Chinese publications, identified by a CSTA order number, are available in Chinese or English translation from International Information Service, Ltd., P.O. Box 24683, ABD Post Office, Hong Kong.

Institution of Mechanical Engineers publications are available in U.S.: SAE Customer Service, Dept. 676, 400 Commonwealth Drive, Warrendale, PA 15096, by quoting the SAE-MEP number.

When ordering, the pertinent order number should always be included, <u>not</u> the **DIGEST** abstract number.

A List of Periodicals Scanned is published in issues, 1, 6, and 12.

MECHANICAL SYSTEMS

ROTATING MACHINES

86-1506
Torsional Vibration Control of Rotating Shaft Systems When Starting and Stopping
K. Nonami, M. Higashi, T. Totani
NASA Lewis Res. Ctr., Cleveland, OH
Bull. JSME, 28 (245), pp 2715-2722 (Nov 1985) 11 figs, 8 refs

KEY WORDS: Shafts, Torsional vibrations, Vibration control

This paper deals with the torsional vibration control problems of a rotor shaft during start-up and shut-down. Two control methods are proposed in this paper. One is an optimal regulator method, and the other is an optimal tracking regulator method. These two control methods consist of constant feedback coefficients and compensation inputs. The authors investigated in detail the two control methods concerning the above mentioned rotor shaft problems. From both the simulations and experiments, it was found that the optimal tracking method was superior to the optimal regulator method.

86-1507
Coupled Torsional-flexural Vibration of a Shaft in a Geared System
Hiroshi Iida, Akiyoshi Tamura, Masataka Oonishi
Tokyo Inst. of Technology, Tokyo, Japan
Bull. JSME, 28 (245), pp 2694-2698 (Nov 1985) 9 figs, 1 table, 2 refs

KEY WORDS: Shafts, Gear drives, Torsional vibrations, Flexural vibrations

A gear train system sometimes consists of more than two shafts. But dynamic characteristics of such systems have not been analyzed in detail. Especially, on a counter shaft on which there are two pairs of gears in mesh, their power transmitting direction does not coincide each other. As a result, vibrations of power transmitting direction and tooth sliding direction couple together even if the gyro effect is ignored. The dynamic characteristics of a counter shaft from this point of view are discussed.

86-1508
A Theory of Post-Stall Transients in Axial Compression Systems: Part I — Development of Equations
F.K. Moore, E.M. Greitzer
Cornell Univ., Ithaca, NY
J. Engrg. Gas Turbines Power Trans. ASME, 108, pp 68-76 (Jan 1986) 7 figs, 10 refs

KEY WORDS: Compressors, Stalling, Fluid-induced excitation

An approximate theory is presented for post-stall transients in multistage axial compression systems. The theory leads to a set of three simultaneous nonlinear third-order partial differential equations for pressure rise, and average and disturbed values of flow coefficient, as functions of time and angle around the compressor. By a Galerkin procedure, angular dependence is averaged, and the equations become first order in time. These final equations are capable of describing the growth and possible decay of a rotating-stall cell during a compressor mass-flow transient. It is shown how rotating-stall-like and surgelike motions are coupled through these equations, and also how the instantaneous compressor pumping characteristic changes during the transient stall process.

86-1509
A Distinction Between Different Types of Stall in a Centrifugal Compressor Stage
N. Kammer, M. Rautenberg
Universität Hannover, Hannover, West Germany
J. Engrg. Gas Turbines Power Trans. ASME, 108, pp 83-92 (Jan 1986) 18 figs, 1 table, 14 refs

KEY WORDS: Centrifugal compressors, Stalling, Fluid-induced excitation

The flow at the stall line of a centrifugal compressor with vaneless diffuser we investigated at different speeds. A distinction between three kinds of stall phenomena could be made. One type of stall with regurgitation of fluid at the impeller inlet was of a nonperiodic character, whereas two different types of periodic stall appeared at higher speeds. The rotating nature of these two types of stall was verified from a comparison of signals of peripherally spaces pressure transducers. The low-frequency rotating stall exhibited features of diffuser generated stall and a lobe number of three was measured. From a detailed investigation of the high-frequency rotating stall, which included unsteady probe measurements upstream and downstream of the impeller, it can be shown that this type of rotating stall is generated in the impeller by a periodic breakdown of energy transfer from the rotor to the flow. This conclusion is supported by the distribution of shroud static pressures.

86-1510
Dynamic Response of a Centrifugal Blower to Periodic Flow Fluctuations
A.N. Abdel-Hamid
The American Univ. in Cairo, Cairo, Egypt
J. Engrg. Gas Turbines Power Trans. ASME, 108, pp 77-82 (Jan 1986) 12 figs, 16 refs

KEY WORDS: Centrifugal compressors, Fluid-induced excitation

Experimental investigation of the dynamic response of a centrifugal blower to periodic flow rate modulations was carried out at a different blower operating conditions. For modulation frequencies in the range of 0.0086-0.085 of the shaft rotation frequency, the fluctuating pressures at inlet, discharge, and across a flow orifice were simultaneously measured and analyzed in the time and frequency domains. Measurements of the amplitude and phase of the transfer function between the blower static pressure rise and the discharge flow rate fluctuations indicated that the quasi-steady approximation should be limited to frequencies lower than 0.02 of the shaft rotation frequency. For the same average flow rate, the static pressure rise progressively lagged the discharge flow rate fluctuations as the frequency was increased. The trend was similar to that of the inertia effects of a fluctuating flow of a pipe. For the same frequency these inertia effects increased as the average flow rate through the blower was decreased. Applications of the results to on-line measurements of the slope of the characteristic curve and improved dynamic modeling of centrifugal compressors and blowers are discussed.

86-1511
Effects of Nonlinear Spring Characteristics on the Dynamic Unstable Region of an Unsymmetrical Rotor
Yukio Ishida, Takashi Ikeda, Toshio Yamamoto, Makoto Hiei
Nagoya Univ., Nagoya-City, Japan
Bull. JSME, 29 (247), pp 200-207 (Jan 1986) 10 figs, 14 refs

KEY WORDS: Rotors, Nonlinear springs, Ball bearings, Unbalanced mass response, Parametric resonance

This paper deals theoretically and experimentally with the forced oscillation in the vicinity of the dynamic unstable region of an unsymmetrical rotor system having nonlinear spring characteristics due to ball bearings. Near this rotating speed of a shaft, an unstable region vanishes and a stable oscillation appears due to the nonlinearity. This oscillation is discussed with attention to the nonlinear components represented by the polar coordinates. It notes an external force due to the unbalance, nonlinearity and the characteristic of parametric resonance, we have investigated the effect of the coexistence of these factors on the vibratory phenomena. It has been compared to oscillation with the super-summed-and-differential harmonic oscillations in a symmetrical shaft system and the dynamic unstable vibration of an unsymmetrical rotor system reported previously.

86-1512
Stability of a Clamped-Free Rotor Partially Filled With Liquid
S.L. Hendricks
Virginia Polytechnic Institute and State Univ., Blacksburg, VA
J. Appl. Mech., Trans. ASME, 53 (1), pp 166-172 (Mar 1986) 5 figs, 8 refs

KEY WORDS: Rotors, Cylinders, Fluid-filled media, Stability

When a flexible rotor is partially filled with liquid, the motion is unstable over some operating range. The extent of this operating range depends on various system parameters such as rotor damping, fluid viscosity, the amount of fluid present, etc. If the rotor is arranged so that is must tilt as it vibrates (as in the clamped-free configuration) then the tilt of the rotor greatly complicates the stability analysis. The source of the complication is that the fluid motion becomes three-dimensional. A three-dimensional stability theory is developed here and applied to a simple clamped-free rotor. The results show that the stability boundaries are influenced by both rotor and fluid "gyroscopic stiffening" effects. Brief experimental results are also reported.

86-1513
Factors That Affect the Fatigue Strength of Power Transmission Shafting and Their Impact on Design
S.H. Loewenthal
NASA Lewis Res. Ctr., Cleveland, Ohio
J. Mech., Transm., Autom. in Des., 108 (1), pp 106-118 (Mar 1986) 9 figs, 2 tables, 25 refs

KEY WORDS: Shafts, Power transmission systems, Fatigue life

A long-standing objective in the design of power transmission shafting is to eliminate excess shaft material without compromising operational reliability. A shaft design method is presented which accounts for variable amplitude loading histories and their influence on limited life designs. The effects of combined bending and torsional loading are considered, along with a number of application factors known to influence the fatigue strength of shafting materials. Among the factors examined are surface condition, size, stress concentration, residual stress, and corrosion fatigue.

86-1514
Effects of Inflow Conditions on Discrete Frequency Noise Generated by Small, Axial Flow Fans
K.B. Washburn
Pennsylvania State Univ., State College, Applied Research Lab.
Rept. No. PSU/ARL/TR-85-001, 149 pp (Sep 1985) AD-A160 579/9/GAR

KEY WORDS: Fans, Noise generation, Sound waves, Wave radiation

Discrete frequency acoustic radiation is generated by subsonic axial flow fans through both steady and unsteady blade loading. Steady loading is a function of pumping requirement, and unsteady loading is generated by spatially periodic inflow distortions. The latter effect is the dominant generation mechanism when small fans are used to cool electronic equipment. Simple and modified inlet baffles, finger guards, and honeycomb flow straighteners are investigated. Design recommendations are offered to minimize discrete tone generation. These include aerodynamic shaping of unavoidable obstructions, a minimum axial distance of 0.3 fan radii for obstructions, a minimum axial distance of 0.3 fan radii for obstructions in the inlet, the avoidance of blockage in lateral inflow and the use of an inlet baffle to smooth inlet distortions. Account is taken of the spatial restrictions of typical installations.

86-1515
The Vibrations of a Textile Machine Rotor with Nonlinear Characteristics
L.J. Cveticanin
Mech. Mach. Theory, 21 (1), pp 29-32 (1986) 4 figs, 5 refs

KEY WORDS: Rotors, Textile looms, Nonlinear systems

In this paper the free vibrations of a textile machine rotor which rotates with constant angular velocity are considered. The rotor consists of a weightless shaft and a disc with variable mass. The force in the shaft is assumed to be nonlinear. Mass of the disc is varying due to the winding up of the textile band. Severe vibrations occur due to the mass varying. When the nonlinearity is small and the variation of the mass is a function of "slow time," the vibrations can be denoted not only numerically but also analytically by use of the multiple scales method.

POWER TRANSMISSION SYSTEMS

86-1516
Nonlinear Response of Roller Chain to Combined Forcing and Parametric Excitations
Atsuo Sueoka, Hideyuki Tamura, Takahiro Kondou, Toshiro Fujimoto
Kyushu Univ., Fukuoka, Japan
Bull. JSME, 29 (247), pp 178-186 (Jan 1986) 4 figs, 12 refs

KEY WORDS: Chains, Forced vibration, Parametric excitation

This paper describes the results of analytical and experimental investigations of the nonlinear harmonic and parametric resonance phenomena of a roller chain. It is stretched vertically. The resonance treated in this report is one of the resonances excited under combined forcing and parametric excitations in which the ratio of the forced lateral displacement acting at a lower end to that of the tension fluctuation is 1:2. Two chains with different frictional coefficients were used in experiments in order to examine the effect of friction upon the vibration characteristics. The experimental results showed remarkable features attributable to the combined excitations and agreed well with the analytical results.

METAL WORKING AND FORMING

86-1517
Application of Multi-dimensional Spectral Analysis for Noise Source Identification on Forge Machine
Jae Eung Oh

Han Yang Univ., Seoul, Korea
Bull. JSME, 28 (245), pp 2744-2748 (Nov 1985) 7 figs, 2 tables, 5 refs

KEY WORDS: Forging machinery, Noise source identification, Spectrum analysis

This paper presents a new method for noise source identification in a multiple noise sources environment where the noise sources may be coherent to each other. It is found that the major characteristics proportion of the noise source is generated in vibration of forge machine by using multi-dimensional spectral analysis. In this analysis the concept of residual spectral analysis and the partial coherence function are applied. For their contributions, relationships between noise sources and radiated sound pressure are computed by micro-computer system. Finally, overall levels for radiated sound pressure obtained by multi-dimensional spectral analysis are compared with those measured.

MATERIALS HANDLING EQUIPMENT

86-1518
A Study on Parametric Vibration in Chuck Work
Masahiro Doi, Masami Masuko, Yoshimi Ito, Atsushi Tezuka
Musashi Inst. of Technology, Tokyo Japan
Bull. JSME, 28 (245), pp 2774-2780 (Nov 1985) 13 figs, 1 table, 8 refs

KEY WORDS: Machine tools, Parametric vibration

In this paper, the parametric instability in chuck work process is investigated. And an analytical criterion of stability is given, and experimental confirmation is carried out. The vibration of this type is considered to have a significant influence on machinability. However, the applicable isolation of this vibration has not been done.

STRUCTURAL SYSTEMS

BRIDGES

86-1519
Analysis on the Longitudinal Coupled Vibration of Railway Bridges and Piers
Zhou Hongye, Ma Changshui
J. China Railway Soc., 7 (3), pp 66-74 (1985)
CSTA No. 625.1-85.28

KEY WORDS: Railroad bridges, Piers

This paper studies the calculation method and its approximation approach for the characteristics of longitudinal coupled vibration of railway bridges and piers. It is pointed out the longitudinal coupled vibration can be approximately summed up as the vibration of a single pier on which a concentrated mass M is applied.

BUILDINGS

86-1520
Modal Time History Analysis of Non-Classically Damped Structures for Seismic Motions
M.P. Singh, M. Ghafory-Ashtiany
Virginia Polytechnic Institute and State Univ., Blacksburg, VA
Earthquake Engrg. Struc. Dynam., 14 (1), pp 133-146 (Jan-Feb 1986) 6 figs, 2 tables, 17 refs

KEY WORDS: Structural response, Seismic response, Mode acceleration method, Mode displacement method, Multistory buildings

The step-by-step modal time history integration methods are developed for dynamic analysis of non-classically damped linear structures subjected to earthquake-induced ground motions. Both the mode displacement and mode acceleration-based algorithms are presented for the calculation of member and acceleration responses. The complex-valued eigenvectors are used to effect the modal decoupling of the equations of motion. However, the recursive step-by-step algorithms are still in terms of real quantities. The numerical results for the acceleration response and floor response spectra, obtained with these approaches, are presented. The mode acceleration approach is observed to be better than the mode displacement approach in as much as it alleviates the so-called missing mass effect, caused by the truncation of modes, very effectively. The utilization of the mode acceleration-based algorithms is, thus, recommended in all dynamic analyses for earthquake-induced ground motions.

86-1521
Dynamic Analysis of Multistory Buildings by Component Mode Synthesis
M.A.M. Torkamani, J.T. Huang

Pittsburgh Univ., Pittsburgh, PA
Rept. No. SETEC-CE-85-008, NSF/CEE-84064, 166 pp (Nov 1984) PB86-106589/GAR

KEY WORDS: Multistory buildings, Component mode synthesis

A combination of reduction procedures, with fixed-interface component mode synthesis as the central theme, augmented by static condensation and Guyan reduction, is formulated and implemented for a six-story structure with a three-dimensional frame. The validity of procedures and the program is established by comparing results to those obtained from SUPERSAP, a general purpose finite element program model. The agreement is shown to be very good. A twelve-story, three-dimensional building with an L-shape floor plan is also analyzed; results indicate that the combined procedures are advantageous in terms of convergence, the structural characteristics preserved, and the percentage of reduction achieved. Results also confirm the importance of floor flexibility.

86-1522
Seismic Response of Brick Buildings with Sliding Substructure
M. Qamaruddin, A.S. Arya, B. Chandra
Univ. of Petroleum and Minerals, Dhahran, Saudi Arabia
ASCE J. Struc. Engrg., 112 (3), pp 558-572 (Mar 1986) 13 figs, 1 table, 27 refs

KEY WORDS: Buildings, Masonry, Seismic response, Seismic isolation, Sliding supports

To investigate the ground motion isolation feasibility, a new brick building system is considered. Pilot tests carried out on one-fourth scale brick building models showed large reductions in the roof acceleration compared to conventional fixed-base models that are subject to shake table motion, indicating a possibility of earthquake isolation. The seismic response of single-story sliding buildings, subjected to Koyna and El Centro accelerograms, is computed through a two-mass spring dashpot mathematical model that treats the frictional resistance as rigid plastic. This study leads us to a concept of frictional response spectra, on which the spectral quantities of the sliding system are plotted against the undamped natural period for various coefficients of friction and mass ratios. These spectra clearly show the reduction of response of the sliding system compared to the conventional building.

86-1523
Measurement of Structural Intensity in Building Constructions
P. Kruppa
Jutland Technological Institute, Denmark
Appl. Acoust., 19 (1), pp 61-74 (1986) 14 figs, 3 refs

KEY WORDS: Buildings, Acoustic insulation, Sound transmission

When designing buildings with a maximum of sound insulation, it is important to know the different sound transmission paths in building structures. A direct measurement of the structure-borne sound intensity has been tried earlier on thin-plate constructions in laboratory set-ups. In this investigation an equivalent technique has been tried on practical building constructions, and the results appear to be promising even on relatively thick and heavy concrete walls. The technique uses normal sound intensity measuring equipment. The experimental work was carried out in a specially designed flanking transmission laboratory.

86-1524
Computer Program for the Analysis of Asymmetric Frame-Shear Wall Structures
S. Swaddiwudhipong, T. Balendra, S.T. Quek, S.L. Lee
National Univ. of Singapore, Kent Ridge, Singapore
Computers Struc., 22 (3), pp 343-362 (1986) 7 figs, 2 tables, 9 refs

KEY WORDS: Buildings, Computer programs, Cantilevers, Galerkin method

A computer program for the analysis of tall buildings comprising frames and shear walls coupled together is presented. Both static and free vibration analyses of the buildings of both uniform and nonuniform sections on either rigid or flexible foundation are considered. The governing equations are formulated through the continuum approach treating the structures as shear-flexure cantilevers. Both polynomial and transcendental displacement functions are employed to approximate the true displacement field. The method is shown to be simple yet powerful.

86-1525
Simplified Procedures for Earthquake Analysis of Buildings
E.F. Cruz, A.K. Chopra

Univ of California, Berkeley, CA
ASCE J. Struc. Engrg., 112 (3), pp 461-480 (Mar 1986) 8 figs, 2 tables, 12 refs

KEY WORDS: Buildings, Seismic analysis, Seismic response spectra

The earthquake response of many buildings can be estimated by considering only the first two vibration modes in the response spectrum analysis (RSA) procedure. A simplified response spectrum analysis (SRSA) procedure is presented. The SRSA method should be very useful in practical application because, although much simpler than the RSA method, it provides very similar estimates of design forces for many buildings. With the development of the SRSA, a hierarchy of four analysis procedures to determine the earthquake forces are available to the building designer: code-type procedure, SRSA, RSA, and RHA (response history analysis). The criteria presented to evaluate the results from each procedure and to decide whether is it necessary to improve results by proceeding to the next procedure in the hierarchy use all the preceding computations and are, therefore, convenient.

86-1526

Elastic Earthquake Response of Building Frames

E.F. Cruz, A.K. Chopra
Univ. of California, Berkeley, CA
ASCE J. Struc. Engrg., 112 (3), pp 443-459 (Mar 1986) 16 figs, 3 tables, 9 refs

KEY WORDS: Buildings, Framed structures, Seismic analysis, Seismic response spectra

The accuracy of the response spectrum analysis (RSA) for estimating the maximum response of a building directly from the earthquake design spectrum is evaluated with the objective of developing better simplified analysis procedures. The procedures can also be included in building codes. For a fixed fundamental period T_1 of the building, the response contributions of the higher vibration modes increase, and consequently, the errors in the RSA results increase, with decreasing beam-to-column stiffness ratio for a fixed stiffness ratio, the response contributions of the higher vibration modes increase. Consequently, the errors in the RSA results increase, with increasing T_1 in the medium- and long-period regions of the design spectrum. The RSA results are accurate enough for design applications. Based on the results presented in this paper, improved simplified analysis procedures for the preliminary design of buildings are developed in the companion paper.

86-1527

Simplified Seismic Analysis of Secondary Systems

R. Villaverde
Univ. of California, Irvine, CA
ASCE J. Struc. Engrg., 112 (3), pp 588-604 (Mar 1986) 5 figs, 9 tables, 19 refs

KEY WORDS: Equipment-structure interaction, Seismic response spectra

A procedure is introduced to compute the maximum response of light secondary systems attached to buildings subjected to earthquakes. The procedure is based on the application of the response spectrum technique to a combined building-attachment system. It is formulated in terms of the natural frequencies and mode shapes of the independent building and attachment components. It takes into account the interaction between the two components, avoids the solution of large eigenvalue problems, and facilitates the analysis of different secondary systems without the need to reanalyze the supporting building. The procedure is formulated by using Rayleigh's principle and Hurty's component mode synthesis technique. Linear systems with classical modes of vibration, small secondary- to primary-mass ratios, and no more than one point of attachment are considered. In a comparative numerical study the proposed procedure predicts correct solutions with an average error of about 1%.

TOWERS

86-1528

Anchorage Requirements for Wind-loaded Empty Silos

D. Briassoulis, D.A. Pecknold
ASCE J. Struc. Engrg., 112 (2), pp 308-325 (Feb 1986) 11 figs, 5 tables, 25 refs

KEY WORDS: Grain silos, Wind-induced excitation, Finite element technique, Cantilever beams

Base anchorage forces for three empty, stiffened steel grain silos with conical roof shells are determined by finite element analysis for nonuniform wind loading. In general, the axial tensile forces govern for tall silos, while the shear force anchorage requirements control for shallow silos. The anchorage requirements for all three types of silos are adequately predicted by use of an adjusted membrane theory for a wind loaded cylindrical shell free at its top, with axial stress at the top due to the roof wind loading. The "cantilever beam" approximation yields extremely nonconservative anchorage forces.

FOUNDATIONS

86-1529

Determination of Parameters for a Model for the Cyclic Behaviour of Interfaces

E.C. Drumm, C.S. Desai
Univ. of Tennessee, Knoxville, TN

Earthquake Engrg. Struc. Dynam., 14 (1), pp 1-18 (Jan-Feb 1986) 18 figs, 1 table, 20 refs

KEY WORDS: Soil structure interaction, Sand, Concrete, Cyclic loading

The cyclic shear stress-deformation response of dry sand-concrete interfaces is described using a modified form of a Ramberg-Osgood (R-O) model. The model permits the description of the interface secant stiffness as a function of the normal stress, shear stress, sand density and number of loading cycles. Stiffening behavior, or an increase in secant shear stiffness with number of cycles, may be represented, as well as degradation behavior. Typical results from a series of cyclic direct shear tests are presented. A methodology is described by which the parameters of the (R-O) model can be determined from the laboratory shear tests, and the functional forms of the parameters for the sand-concrete interface are provided. The model is verified by predicting the response of two displacement controlled laboratory tests that were used for the determination of the model parameters.

86-1530

Dynamic Finite Element Analysis of Three-Dimensional Soil Models with a Transmitting Element

H. Werkle
Hochtief AG, Frankfurt, W. Germany

Earthquake Engrg. Struc. Dynam., 14 (1), pp 41-60, (Jan-Feb 1986) 15 figs, 1 table, 19 refs

KEY WORDS: Soil-structure interaction, Finite element technique

A method for the dynamic finite element analysis of a non-axisymmetric soil model with an axisymmetric boundary is presented. In the non-axisymmetric soil domain, an arbitrary discretization with three-dimensional isoparametric solid elements is used. At the boundary a transmitting element is arranged. It is based on the semi-analytical element of Waas and Kausel. The transformation of the stiffness matrix of the Waas/Kausel element with cyclic symmetric displacements to general displacement field is presented. For earthquake excitation the forces acting on the discretized domain are given. The method is illustrated by the dynamic analysis of an embedded box-type building. The distribution and magnitude of significant section forces are discussed.

86-1531

A Hybrid Method for Three-Dimensional Problems of Dynamics of Foundations

Hong-Tsung Lin, J.L. Tassoulas
Univ. of Texas, Austin, TX

Earthquake Engrg. Struc. Dynam., 14 (1), pp 61-74 (Jan-Feb 1986) 7 figs, 3 tables, 10 refs

KEY WORDS: Soil-structure interaction, Foundations, Finite element technique

A method is developed applicable to problems of dynamics of arbitrary-geometry foundations. Finite elements are employed in the near field in order to obtain a discrete solution. In the far field, a semidiscrete solution is synthesized from modes also calculated by the finite element method. The solutions are matched by applying the stationarity condition of a functional. Examples of application are presented in order to verify the validity and illustrate the use of the method.

86-1532

Non-Linear Soil-Structure Interaction Analysis Based on the Boundary-Element Method in Time Domain with Application to Embedded Foundation

J.P. Wolf, G.R. Darbre
Electrowatt Engineering Services Ltd., Zürich, Switzerland

Earthquake Engrg. Struc. Dynam., 14 (1), pp 83-101 (Jan-Feb 1986) 10 figs, 1 table, 16 refs

KEY WORDS: Soil-structure interaction, Foundations, Time domain method, Boundary element technique

The various boundary-element methods, well established in the frequency domain, are developed in the time domain for a foundation embedded in a layered halfspace. They are the weighted-residual technique and the indirect

boundary-element method, based on a weighted-residual equation, and the direct boundary-element method. As an example, the nonlinear soil-structure interaction analysis of a structure embedded in a halfspace with partial uplift of the basemat and separation of the side wall is investigated.

86-1533
Structural Response for Six Correlated Earthquake Components
M. Ghafory-Ashtiany, M.P. Singh
Univ. of Gilan, Iran
Earthquake Engrg. Struc. Dynam., 14 (1), pp 103-119 (Jan-Feb 1986) 1 fig, 7 tables, 21 refs

KEY WORDS: Seismic response, Structural response, Modal analysis

The paper examines the effect on the structural response of the inevitable correlation which exists between the six earthquake components acting along a set of structural axes. The rotational components are expressed in terms of the spatial derivatives of the translational components. For the calculation of response, modal analysis is employed so that ground response spectra can also be used as seismic input. A methodology for the calculation of design response is advocated, especially for asymmetric structures. For the assumed model of seismic wave motion, the numerical results show a significant contribution to the response from the rotational components. This contribution is, however, expected to be reduced by structural foundation averaging and interaction effects. Further studies with more complete models of seismic wave motions, and their interaction with structural foundations, are thus warranted for a realistic evaluation and characterization of the rotational inputs for design purposes.

HARBORS AND DAMS

86-1534
Seismic Shear Vibration of Embankment Dams in Semi-Cylindrical Valleys
P. Dakoulas, G. Gazetas
Rensselaer Polytechnic Institute, Troy, NY
Earthquake Engrg. Struc. Dynam., 14 (1), pp 19-40 (Jan-Feb 1986) 13 figs, 27 refs

KEY WORDS: Dams, Seismic response

A rigorous analytical solution is developed for the lateral linear shear response of embankment dams and semi-cylindrical valleys. Closed-form algebraic expressions are presented pertaining to both free and base-induced oscillations and extensive parametric and comparative studies elucidate prominent effects of canyon geometry (shape and aspect ratio) on dynamic response. Harmonic steady-state as well as earthquake-induced accelerations, displacements and shear strains in the dam are studies and compared with those obtained from three dimensional analyses for other canyon geometries, as well as from two dimensional analyses of the dam mid-section. It is shown that such two dimensional analyses may provide significantly lower values of near-crest accelerations, but slightly higher values of shear strains and stresses that the three dimensional analyses. The proposed method of analysis is at least three orders of magnitude less expensive than other presently available numerical procedures.

POWER PLANTS

86-1535
On the Vibration of Boiler Structures in Large Power Plants
Zhu Jimei
JSIME (3), pp 1-16 (1985) CSTA No. 621.8-85.54

KEY WORDS: Power plants, Boilers

This article describes the comprehensive measurements made on the vibrations of the furnace walls and the attached back stays on the boiler plants. The measured data are processed in order to get the power spectrum of the flue gas pressure and the welded tube wall's vibration. Moreover, the transfer function between them is analyzed to detect the dynamical characteristics of the system and the randomness of the vibrations due to burning. Karman vortices and sound resonance in the furnace chamber are also qualitatively examined.

86-1536
Aseismic Study of High Temperature Gas-cooled Reactor Core with Block-type Fuel
T. Ikushima, T. Honma
Japan Atomic Energy Res. Inst., Ibaraki-ken, Japan
Bull. JSME, 28 (246), pp 2986-2993 (Dec 1985) 17 figs, 3 tables, 5 refs

KEY WORDS: Nuclear reactors, Seismic tests

A two-dimensional horizontal seismic experiment with single axis and simultaneous two-axes excitations was performed to obtain the core seismic design data on the block-type high temperature gas-cooled reactor. Effects of excitation directions and core side support stiffness on characteristics of core displacements and reaction forces of support were revealed. The values of the side reaction forces are the largest in the excitation of flat-to-flat of hexagonal block. Preload from the core periphery to the core center are effective to decrease core displacements and side reaction forces.

VEHICLE SYSTEMS

GROUND VEHICLES

86-1537

Analysis of the Nonstationary Response of Vehicles with Multiple Wheels

R.F. Harrison, J.K. Hammond
Oxford Univ., Oxford, England

J. Dynam. Syst., Meas. Control, Trans. ASME, 108 (1), pp 69-73 (Mar 1986) 5 figs, 9 refs

KEY WORDS: Ground vehicles, Bicycles, Surface roughness

Vehicles moving on rough surfaces are subject to inputs which are often conveniently regarded as random processes. In general, the excitation process is perceived by the vehicle as a nonstationary random process either due to inhomogeneity in the ground profile or variations in the vehicle's velocity, or both. Hitherto this second case has not been tractable analytically due to the time variable delay between inputs. In this paper this difficulty is overcome and expressions are derived for the propagation of the mean vector and zero-lag autocovariance matrix. An example of a vehicle modeled by a bicycle configuration is discussed.

86-1538

Research in Rail Vehicle Dynamics — State of the Art

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Concordia Univ., Montreal, Quebec, Canada

Shock Vib. Dig., 18 (2), pp 9-18 (Feb 1986)

KEY WORDS: Railroad trains, Rail-vehicle interaction, Reviews

A survey of the state of the art of research investigations in railway dynamics is presented. Emphasis is given to evaluating the dynamic response of rail vehicles subject to excitations arising from rail input. The nature of rail vehicles and the dynamic loads they experience are described. A review of rail vehicle models employing deterministic and stochastic forms of excitation due to track input is presented. Vehicles and trucks are generally modeled using linear and nonlinear formulations for response determination and stability evaluation. Methods for solving such mathematical models as nonlinear systems under stochastic excitation are briefly described. Optimization techniques utilized in the design solution of the problems are reviewed.

86-1539

Coupled Vibrations between Railway Vehicle Wheels and a Rail

Takashi Ayabe, Atsuo Sueoka, Hideyuki Tamura
Kyushu Univ., Fukuoka, Japan

Bull. JSME, 29 (247), pp 194-199 (Jan 1986) 5 figs, 8 refs

KEY WORDS: Rail-wheel interaction, Periodic excitation

Rolling noise arises from the vibration due to the interaction between wheels and rails through the contact points. The authors treated the vibration of a rail in the horizontal and vertical directions in the previous papers. In this report, the axial vibration of a rotating wheel, at the circumference of which shear force and spin moment from Kalker's linear creep theory act simultaneously, is formulated in order to consider the coupled vibration between wheels and rail in the horizontal direction. It is shown that the response of the steady-state vibration is obtained by solving the simultaneous algebraic equations with respect to the contact forces. As a verification of the present analytical method, the theoretical and experimental impedance characteristics of a stationary wheel suspended with springs at a point on the outer circumference are compared.

SHIPS

86-1540

Solution of the Problem of Ship Towing by Elastic Rope Using Perturbation

M.M. Bernitsas, N.S. Kekridis, F.A. Papoulias

Univ. of Michigan, Ann Arbor, MI
J. Ship Res., 30 (1), pp 51-68 (Mar 1986) 13 figs, 1 table, 17 refs

KEY WORDS: Ships, Towed systems, Perturbation theory

A nonlinear time-dependent system is used to model the horizontal plane motions of a ship towed by a nonlinear elastic rope. The solution of the system by simulatir does not reveal the nature of the various motion components and is expensive because it requires a very small integration step dictated by the rope dynamics. A method is developed, based on perturbation, which decouples the fast towrope tension component from the slow one and reveals the nature of both and their effects on the fast- and slow-motion components. The method is general and can be used for other towline dynamic models as well. The reconstruction of the complete motion based on the results of the three problems, which are derived by perturbation from the original model, gives satisfactory results and verifies the validity of the assumptions used in the decomposition of the original problem.

AIRCRAFT

86-1541

Laboratory Study of Cabin Acoustic Treatments Installed in an Aircraft Fuselage

K.E. Heitman, J.S. Mixson
NASA Langley Res. Ctr., Hampton, VA
J. Aircraft, 23 (1), pp 32-38 (Jan 1986) 18 figs, 9 refs

KEY WORDS: Aircraft fuselages, Interior noise, Noise reduction

The insertion loss of side-wall add-on acoustic treatments was measured using a light aircraft fuselage. The treatments included: no treatment (i.e., baseline fuselage), a production-type double-wall interior, and various amounts of high-density fiberglass added to the baseline fuselage. The source used to simulate propeller noise was a pneumatic-driver with attached exponential horn, supplied with a broadband signal. Data were acquired at the approximate head location for each of the six possible passenger positions. Insertion loss results for the different configurations are analyzed in space-averaged narrowband levels, one-third-octave band levels, and overall levels, and at specific frequencies representing propeller tone spectra. The propeller tone data include not only the space-averaged insertion loss but also the variation of insertion loss of these particular frequencies across the six microphone positions.

86-1542

Control of Aeroelastic Instabilities Through Stiffness Cross-Coupling

T.A. Weisshaar, R.J. Ryan
Purdue Univ., West Lafayette, IN
J. Aircraft, 23 (2), pp 148-155 (Feb 1986) 12 figs, 9 refs

KEY WORDS: Aircraft, Flutter, Stiffness effects

An idealized aeroelastic tailoring model is developed to assess the effects of significant changes in directional stiffness orientation upon the flutter and divergence behavior of swept and unswept wings. A nondimensional stiffness cross-coupling parameter is used to illustrate the potentially strong influence of stiffness cross-coupling, commonly present in aeroelastically tailored structures, to increase flutter and divergence speeds. Conflicting requirements for flutter and divergence enhancement are indicated. Aeroelastic tailoring for flutter enhancement appears to be less effective when the wing is moderately swept back. However, by combining directional stiffness orientation with inertia balancing, flutter and divergence-free, aft-swept, high-aspect-ratio surfaces are shown to be theoretically possible.

86-1543

The Use of High-Strength Steel in Passenger Cars — Part 3: Stiffness, Vibration Characteristics, and Acoustics (Einsatz von höherfesten Stählen in Personenkraftwagen — Teil 3: Steifigkeit, Schwingungsverhalten und Akustik)

H. Krauss, D. Roesems
Automobiltech. Z., 87 (12), pp 683-690 (Dec 1985) 17 figs, 9 refs (in German)

KEY WORDS: Automobile bodies, Stiffness coefficients, Structure borne noise

This part of a research report assesses the possibilities and limits of high-strength steel in cars, the consequences of material substitution and sheet-thickness reduction on the statical and dynamical stiffness. Likewise, the acoustical implications with regard to air-borne noise as well as to structure-borne noise are checked. The measurements are performed on representative components and on complete Porsche 928 bodies and road-going cars.

86-1544
A Propeller Model for Studying Trace Velocity Effects on Interior Noise
J.R. Mahan, C.R. Fuller
Virginia Polytechnic Institute and State Univ., Blacksburg, VA
J. Aircraft, 23 (2), pp 142-147 (Feb 1986) 8 figs, 1 table, 6 refs

KEY WORDS: Aircraft noise, Interior noise, Noise reduction

There is concern that advanced turboprop (ATP) engines currently being developed as an alternative to turbofan engines may produce excessive aircraft cabin noise levels. This concern has stimulated renewed interest in developing aircraft interior noise reduction methods that do not significantly increase takeoff weight. Both syncrophasing and active control of interior noise have been proposed as solutions, but neither has been perfected, mostly from lack of physical understanding of the sound transmission mechanism. This paper exploits an existing analytical model for noise transmission into aircraft cabins to investigate the behavior of an improved propeller source model for use in aircraft interior noise studies.

86-1545
Interactive Effects of High- and Low Frequency Loading on Fatigue
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Mechanical Technology, Inc., Latham, NY
Rept. No. MTI-85TR48, AFWAL-TR-85-4045, 158 pp (May 1985) AD-A160 601/1/GAR

KEY WORDS: Disks, Aircraft engines, Fatigue life

This report describes the results of a program to measure and model the controlling mechanisms of fatigue and creep-crack growth behavior of a typical aircraft engine disk material under high frequency/low frequency loading cycles. The goal of the program is to provide a basis for damage-tolerant design of aircraft engine components under combined high and low frequency loading.

MISSILES AND SPACECRAFT

86-1546
Dynamic Response and Collapse of Slender Guyed Booms for Space Application
J.M. Housner, W.K. Belvin
NASA Langley Res. Ctr., Hampton, VA
J. Spacecraft Rockets, 23 (1), pp 88-95 (Jan/Feb 1986) 14 figs, 2 tables, 7 refs

KEY WORDS: Spacecraft, Antennas, Guyed structures

An analytical and experimental investigation of the nonlinear transient response of eccentric guyed slender booms with tip mass has been performed. The behavior of an experimental model is shown to correlate well with the predicted responses. The analysis has been used to study the transient response associated with slewing maneuvers of such structures in space and to develop an appropriate scaling law for model testing. Both applied step loads of finite duration and imposed initial velocities have been used to approximate transient conditions. Nonlinearities arising from cable slackening, beam-column behavior, and large geometry changes have been incorporated. Dynamic buckling of the boom is possible under excitations that result in cable slackening. Sensitivity to boom lateral eccentricities only when these eccentricities exceed certain threshold values. The nonlinear analysis also has been used to establish design guidelines for combinations of pulse level and duration that meet allowable deflection performance requirements.

86-1547
Flexibility Control of Solar Battery Paddles (1st Report: A Method of Vibration and Attitude Control Based on Outputs of Solar Instrument Sensors)
Toshio Fukuda, Yutaka Kuribayashi, Hidemi Hosokai, Nobuyuki Yajima
Science Univ. of Tokyo, Tokyo, Japan
Bull. JSME, 29 (247), pp 208-213 (Jan 1986) 10 figs, 7 refs

KEY WORDS: Spacecraft, Solar energy, Vibration control

Flexible solar battery paddles of spacecrafts have low frequency vibrational characteristics due to the stringent limitation of the weight of launched rockets. The basic problem dealt with here is how to estimate and control the vibrational modes of flexible booms of the arrays even in large angle attitude maneuvers. First a proposed mode estimation method with use of outputs of solar cells is shown to give good estimation of the vibrational modes. It is shown that even static output maximization control in a desired direction cannot work stably without flexibility control based on the mode estimation and that the boom is controlled dynamically based on the feedback control theory, so as to suppress the vibration of the arrays even in large angle attitude maneuvers.

BIOLOGICAL SYSTEMS

HUMAN

86-1548
A Study of Vibration White Finger Disease Rock Drillers
R.L. Brubaker, C.J.G. Mackenzie, S.G. Hutton
Univ. of British Columbia, B.C., Canada
J. Low Frequency Noise Vib., 4 (2), pp 52-65 (1985) 2 figs, 8 tables, 39 refs

KEY WORDS: Human hand, Human response, Vibratory tools, Drilling, Rocks

Based on a professionally administered medical questionnaire, 50% of 95, rock drillers using hand-held pneumatic drills from two large British Columbia underground mines reported symptoms of Vibration White Finger Disease (VWFD). Prevalence of this disease was 45% among a subgroup of 58 drilled without a medical or occupational history of possible predisposing factors other than drill vibration. Symptoms appeared to be dose-related with blanching attacks reported among 25% of subgroup drillers exposed for 1-5 years, and in 80% of those exposed for 16 years or more. 9% of subgroup drillers had severe stage 3 symptoms (Taylor-Pelmear Classification). The median latent period for onset of blanching symptoms was 7.5 years. Prevalence of VWFD among 58 control miners from the two sites without a history of intense prolonged hand-arm vibration or other predisposing factors was 4%. There was objective evidence of vascular abnormality (based on delayed finger rewarming after combined digital cooling and ischemia) in 76% of subgroup drillers with blanching symptoms and in 18% of controls without symptoms. Comparison is made between the latent period for onset of blanching symptoms among drillers in this study and the predicted latent period based on suggested International Standards Organization guidelines for vibration exposure.

86-1549
Vibration Characteristics of Rock Drills
S.G. Hutton, R.L. Brubaker
Univ. of British Columbia, Vancouver, B.C., Canada
J. Low Frequency Noise Vib., 4 (2), pp 66-80 (1985) 17 figs, 4 refs

KEY WORDS: Human hand, Human response, Vibratory tools, Drilling, Rocks

This report presents the results of field tests conducted to ascertain the vibration characteristics of rock drills from the viewpoint of hand transmitted vibration. Measurements were conducted on a number of jack leg and stoper drills while the drills were being used in a production environment. R.M.S. acceleration spectra are presented and the effects of operating variables are considered.

MECHANICAL COMPONENTS

ABSORBERS AND ISOLATORS

86-1550
Vibration Isolation of a Microphone
C.D. Steele
Naval Postgraduate School, Monterey, CA
Master's Thesis, 93 pp (Sept 1985) AD-A161 018/7/GAR

KEY WORDS: Vibration isolators, Microphones

A microphone vibration isolation system using a bungee elastic suspension, designed for use in a system to measure the ambient noise in the Space Shuttle's payload bay during launch is described. The isolator's transmissibility was measured using a computer controlled shaker table system programmed to simulate the Shuttle's vibrational spectrum in 21 third octave-bands between 20 and 2000 Hertz. Static deflection and transient response measurements verified the axial and radial transmissibility measurements. Free decay measurements were made at 5, 20, and 65C. The isolator's natural frequency of 15 Hertz represents a substantial improvement over the isolator used previously whose lowest resonance was above 100 Hertz. Test procedures and calibration data for three microphones are included.

86-1551
Elastic Coupling of Viscous Torsional Dampers (Zur elastischen Ankopplung von Viskose-Drehschwingungsdämpfern
V. Pistek

Technische Universität Brno, CSSR
Maschinenbautechnik, 34 (12), pp 546-550 (1985) 12 figs, 7 refs (in German)

KEY WORDS: Vibration absorption (equipment), Torsional vibrations, Viscous damping

A theoretical analysis of a vibration absorber coupled in a series arrangement by means of a viscoelastic model is used to illustrate the function of a silicone absorber elastically attached to the housing. The results are verified by experiment.

86-1552
Use of Two Dynamic Vibration Absorbers in the Case of Machinery Operating at Two Frequencies which Coincide with Two Natural Frequencies of the System
J.L. Pombo, P.A.A. Laura
Institute of Applied Mechanics, Puerto Belgrano Naval Base, Argentina
Appl. Acoust., 19 (1), pp 42-45 (1986) 6 figs, 3 refs

KEY WORDS: Dynamic vibration absorption (equipment), Machinery vibration

This paper described an experimental investigation on the use of two dynamic vibration absorbers. It appears, at this point, that the two dynamic absorbers solution is a promising one for this very important and troublesome technological situation.

86-1553
A Primer on Engine Mounts
E.I. Rivin
Dept. of Mech. Engrg., Wayne State Univ., Detroit, MI
Automotive Engrg., 94 (2), pp 77-80, 82, 84-86, 88 (Feb 1986) 3 figs

KEY WORDS: Engine mounts, Automobile engines

This paper deals with the optimization of engine suspension systems in automobiles. They can be greatly improved by passive mount refinements, without resorting to more costly and complex active systems.

86-1554
On the Application of Optimum Damped Absorber to Vehicle Suspension
H. Ghoneim, R.A. Cheema
Rochester Institute of Technology, Rochester, NY
J. Mech., Transm., Autom. in Des., 108 (1), pp 22-24 (Mar 1986) 4 figs, 6 refs

KEY WORDS: Suspension systems (vehicles), Damper locations, Absorbers (equipment)

In a previous paper the advantage of the application of optimum damped absorbers for vehicle suspensions was demonstrated. In that paper the performance characteristics of vehicle suspensions with optimum damped absorbers were compared with those of the pertinent optimized conventional and optimal active systems. Herein the effect of the location of the damped absorbers is studies; i.e., attached to the sprung or unsprung mass. Adopted is a one-dimensional two-degree-of-freedom linear model subjected to random guideway disturbance. Optimization is based on the rms tire-terrain normal force (as a measure of wheel controllability) constrained by the rms sprung mass acceleration (as a measure of the ride comfort). The power spectral density (PSD) of the sprung mass acceleration is evaluated to provide for an additional vehicle performance.

86-1555
Optimal Design of Impact Absorber for Machine-Floor System Under Impact Loads
Y.Z. Wang
National Tsing Hua Univ., Hsinchu, Taiwan, Rep. of China
Appl. Acoust., 19 (3), pp 183-202 (1986) 9 figs, 7 refs

KEY WORDS: Absorbers (equipment), Machine foundations, Floors, Shock absorbers

The optimal design of an impact absorber for a machine mounted on a floor system is described. The floor is considered to be a plate-like structure. A closed-form function is derived. Based on the pulse force excitation, the optimal tuning and damping ratios of the absorber are determined by minimizing the mean squared displacement (velocity, acceleration) response of the machine. The steepest descent method is used to decide these optimal parameters. The effects of mass ratios, i.e. absorber/machine and machine/floor, primary damping, the frequency ratio and the shock duration on the design parameters are discussed .The results show that is is better to keep the duration of the nondimensional shock as long as possible.

SPRINGS

86-1556

Modelling the Ends of Compression Helical Springs for Vibration Calculations

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Univ. of Aberdeen, UK

IMechE. Proc., Part C: Mech. Engrg. Sci., 200 (C1), pp 1-11 (1986) 5 figs, 4 tables, 20 refs

KEY WORDS: Helical springs, Natural frequencies

The squared ends of compression helical springs are modeled as fixed ends with adjacent pinned supports. A theory is developed for the calculation of the natural frequencies, its predictions being compared with experiments on springs with about 2 1/2, 4 and 8 1/2 active turns, using two new designs of test rig. The discrepancy between theory and experiment is typically 1 per cent. This paper contains discussions of the application of the theory to forced vibration calculations, and the modeling of other types of end. Two methods are reviewed for the design of helical springs with the object of avoiding vibration problems.

BLADES

86-1557

The Forced Response of Shrouded Fan Stages

C.-H. Menq, J.H. Griffin, J. Bielak

Carnegie-Mellon Univ., Pittsburg, PA

J. Vib. Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 50-55 (Jan 1986) 7 figs, 15 refs

KEY WORDS: Shrouds, Fan blades, Bladed disks

This paper presents a general approach for modeling shrouded blade vibration that takes into consideration the nonlinear friction constraint at the shroud interface. In this approach, linear structures are characterized by receptances and shroud constraints by nonlinear impedances. The proposed methodology is presented in detail for simplified models of the bladed disk and shroud interface. As an example the method is applied to an idealized tuned stage. Two cases are considered, a lubricated shroud for which the coefficient of friction is equal to zero, and a frictionally constrained shroud. The effect of varying the shroud-to-shroud preload is studies. In the lubricated case nonlinear behavior is seen when vibrations are strong enough to result in separation of the shroud interfaces. In the case of finite friction there is a profound change in resonant frequencies when the preload is increased sufficiently to prevent gross slip at the shrouds.

86-1558

An Experimental Investigation of Blade-Vortex Interaction at Normal Incidence

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Bolt Beranek and Newman, Inc., Cambridge, MA

J. Aircraft, 23 (1), pp 47-55 (Jan 1986) 14 figs, 2 tables, 21 refs

KEY WORDS: Blades, Vortex-induced excitation, Sound waves, Wave radiation, Experimental data

Blade-vortex interaction (BVI) near and away from the blade tip was experimentally investigated for normal incidence where the vortex is generally parallel to the rotor axis. The experiment was designed to be representative of the chopping of helicopter main rotor tip vortices by the tall rotor. Tip Mach number, radial BVI station, and freestream velocity were varied. Fluctuating blade pressures, far-field sound pressure level and directivity, velocity field of the vortex, and BVI angles were measured. BVI away from the tip was that of a dipole, with the direction of minimum radiation, previously found to be in the plane of the blade, rotated considerably in the direction of negative angle of attack. This is believed to be due to unsteady drag radiation caused by stronger BVIs in the present study. For BVI near the tip, the intensity of the interaction was reduced and the radiation pattern was more complex. Away from the tip, the peak amplitude of fluctuating pressure on the suction side of the blade (near the leading edge) was larger than that on the pressure side, as expected due to the effect of compressibility. This trend was reversed near the tip.

86-1559

Influence of Nonlinear Blade Damping on Helicopter Ground Resonance Instability

D.M. Tang, E.H. Dowell

Duke Univ., Durham, NC

J. Aircraft, 23 (2), pp 104-110 (Feb 1986) 13 figs, 7 refs

KEY WORDS: Propeller blades, Helicopters, Ground resonance, Nonlinear damping

The lagging motion of each helicopter blade is assumed to be of equal amplitude and equally apportioned phase, thus allowing a simplified analytical method to be used to calculate the ground resonance instability of a helicopter model with nonlinear dampers in both the landing gear and blades. The geometrical nonlinearities of the blade lag motion and the influence of initial disturbances on ground resonance instability are also discussed. Finally, an experiment is carried out using a helicopter scale model. The experimental data agree well with analysis.

86-1560
Aeroacoustics of an Advanced Propeller Design Under Takeoff and Landing Conditions
S. Fujii, H. Nishiwaki, K. Takeda
National Aerospace Lab., Tokyo, Japan
J. Aircraft, 23 (2), pp 136-141 (Feb 1986) 18 figs, 1 table, 10 refs

KEY WORDS: Propeller blades, Aircraft propellers, Noise generation, Experimental data

Three configurations of six-bladed, 400-mm-diameter scale-model advanced propellers such as backward-, forward-, and back-forward alternately installed swept blades were tested in anechoic environments with an incoming main flow velocity up to 68 m/s. The data for the advance ratio of 0.43-1.15 were obtained in the aeroacoustic aspects. The arrangement of alternately swept blades showed the best quality among the three configurations in the spanwise circulation distribution. The forward-swept blades did not exhibit any aeroacoustic advantage. The alternately swept configuration as a tandem rotation has the potential for decreasing the sound levels at the blade passage frequencies by the dispersion of sound with neither any sacrifice of aerodynamic performance, nor mechanical complexity.

86-1561
A Time Marching Method for Calculating Unsteady Flow Fields Around Vibrating Cascades
Zikang Jiang, Weiwei Zhang
J. Engrg. Thermophysics, 6 (3), pp 237-243 (1985) CSTA No. 621.43-85.104

KEY WORDS: Cascades, Fluid-induced excitation

In this paper stable nonsteady flow fields around vibrating cascades are calculated by use of a time marching method. Vibrating meshes are adopted. In order to reduce the calculating time, the steady flow field derived with a time marching method is used as the initial field, and the calculating meshes vibrates with the cascade. Time marching is continued until the calculated parameters vary with time periodically with stable amplitudes and phases of vibration. Three calculating examples are presented and one of them is compared with the experimental values of a similar cascade showing the effectiveness of this calculating method.

86-1562
An Experimental Investigation into Unsteady Blade Forces and Blade Losses in Axial Compressor Blade Cascade
F. Sugeng, K. Fiedler
Hochschule der Bundeswehr, Hamburg, Fed. Rep. Germany
J. Engrg. Gas Turbines Power Trans. ASME, 108 (1), pp 47-52 (Jan 1986) 13 figs, 9 refs

KEY WORDS: Compressor blades, Fluid-induced excitation, Wind tunnel testing, Blade loss dynamics, Cascades

The unsteady flow problem in the axial compressor has been simulated in wind-tunnels by means of high-speed rotating cylinders upstream of the blade row. To obtain the dynamic changes in the flow properties 20 high-response pressure transducers and 20 high-response hot-film, which are embedded at the surface of the selected blade, have been used in connection with a periodic sampling and averaging technique in digital data acquisition and reduction. The fluctuating forces and the fluctuating drags acting on the blade due to passing wakes shedding through the blade row are carried out.

86-1563
Unsteady Pressure Measurements on a Biconvex Airfoil in a Transonic Oscillating Cascade
L.M. Shaw, D.R. Boldman, A.E. Buggele, D.H. Buffum
NASA Lewis Res. Ctr., Cleveland, OH
J. Engrg. Gas Turbines Power Trans. ASME, 108 (1), pp 53-59 (Jan 1986) 12 figs, 9 refs

KEY WORDS: Flutter, Airfoils, Blades, Cascades, Experimental data

Flush-mounted dynamic pressure transducers were installed on the center airfoil of a transonic oscillating cascade to measure the unsteady aerodynamic response as nine airfoils were simultaneously driven to provide 1.2 deg of pitching motion about the midchord. Initial tests were performed at an incidence angle of

0.0 deg and a Mach number of 0.65 in order to obtain results in a shock-free compressible flow field. Subsequent tests were performed at an angle of attach of 7.0 deg and a Mach number of 0.80 in order to observe the surface pressure response with an oscillating shock near the leading edge of the airfoil. Results are presented for interblade phase angles of 90 and -90 deg and at blade oscillatory frequencies of 200 and 500 Hz (semichord reduced frequencies up to about 0.5 at a Mach number of 0.80). Results from the zero-incidence cascade are compared with a classical unsteady flat-plate analysis. Flow visualization results depicting the shock motion on the airfoils in the high-incidence cascade are discussed. The airfoil pressure data are tabulated.

BEARINGS

86-1564

Characteristics of a Hybrid Journal Bearing with One Recess — Part 1: Dynamic Considerations

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Trib. Intl., 18 (6), pp 331-339 (Dec 1985) 21 figs, 10 refs

KEY WORDS: Journal bearings, Fluid-film bearings

A fluid film hybrid journal bearing with one recess, as used in tandem cold rolling mills, is studied theoretically in two ways. The dynamic response of the hybrid bearing, under isothermal conditions due to the decrease in hydrostatic pressure, is considered and presented in Part 1 of the study. The thermal effects on the load capacity temperature distribution of the bearing will be dealt with and described in Part 2. In this paper, the dynamic behavior of the journal due to the decrease in hydrostatic pressure is presented in the form of transient orbits and squeezed-film speeds. They are shown to be dependent on the initial equilibrium conditions. In the analysis, when the recess pressure is dropped below the hydrodynamic pressure generated by the fluid film, it is found to be difficult to obtain a convergent solution. The dynamic response of the bearing, due to the shut-off of external pressure is, therefore, simulated by the dynamic behavior of the journal due to a series of pressure drops in arbitrary time intervals. The results show that the journal is quite stable in such conditions.

86-1565

Optimization of Tilting Pad Journal Bearings Including Turbulence and Thermal Effects

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Israel J. Tech., 22 (2/3), pp 143-154 (1984/85) 12 figs, 5 tables, 15 refs

KEY WORDS: Journal bearings, Tilt pad bearings, Spring constants, Damping coefficients, Turbulence

The paper presents a comprehensive analysis of tilting pad journal bearings which includes turbulence and thermal effects, and in which most parameters such as pad arc, pivot position, number of pads, preload, direction of load, and others, are arbitrary inputs to the computer solution. The program also calculates spring and damping coefficients under the particular conditions of both journal and pad perturbations. Following a parametric study for the effects of pad arc, preload, pivot position, and mode of loading an optimized geometry is derived which minimizes power loss to load ratio over the usual range of bearing operation. Performance data for the entire range of eccentricities are then given for the optimized bearing.

GEARS

86-1566

Influence of Gear Error on Rotational Vibration of Power Transmission Spur Gear

K. Umezawa, T. Sato

Tokyo Inst. of Technology, Yokohama, Japan

Bull. JSME, 28 (246), pp 3018-3024 (Dec 1985) 11 figs, 1 table, 8 refs

KEY WORDS: Spur gears, Power transmission systems

In this study a power transmission spur gear is recommended as a profile corrected spur gear. Supposing it is allowable that the given value deviates about ten-odd percent from actual acceleration, it is synthesized how an accumulative pitch error influences the rotational vibration of a profile corrected spur gear, using the simulator developed by authors.

86-1567

Prediction Method of Gear Noise Considering the Influence of the Tooth Flank Finishing Method

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J. Vib. Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 95-100 (Jan 1986) 13 figs, 10 refs

KEY WORDS: Gear noise, Noise prediction, Spur gears, Helical gears

Simple and accurate prediction methods of gear unit noise have been desired. This paper offers a new prediction equation for spur and helical gears under speed reduction service. A semi-empirical equation was developed by means of the addition of a dynamics term to Kato's equation which represented the overall noise level by using gear's specification data. As the proposed method takes into account the gear error characteristics in the vibration analysis of gear pairs, it can calculate the noise levels considering the influence of the tooth flank finishing method. It is shown that the predicted values agree with the measured values in an experiment within a range of approximately 5 dB, under almost all the operating conditions and for all test gears having different finishing methods such as hobbing, Niles-type grinding and Maag-type grinding. Moreover, good agreement with data of some actual gear units indicates that the developed method may be used for general application.

86-1568
Dynamic Behavior of Thin-rimmed Spur Gears with Various Web Arrangements
S. Oda, K. Miyachika, T. Koide, T. Fujii
Tottori Univ., Tottori, Japan
Bull. JSME, 29 (247), pp 241-248 (Jan 1986) 11 figs, 7 refs

KEY WORDS: Spur gears, Flexural vibrations, Natural frequencies

This paper presents a study on dynamic behavior of thin-rimmed spur gears with various web arrangements. Natural frequencies of flexural vibration of gear body were measured and compared with the calculated results by Mindlin's method. The relations between these natural frequencies and the spectra of circumferential, radial and axial vibrations were investigated. The circumferential, radial and axial vibration accelerations and root stresses were measured under different running conditions by using a gear testing machine of power absorbing type. On the basis of these results the effects of web arrangements on vibration and dynamic load were clarified to a considerable extent.

86-1569
Vibration of Planetary Gears (Schwingungsverhalten von Planetengetrieben)
H. Peeken, C. Troeder, G. Antony
Konstruktion, 37 (11), pp 417-421 (Nov 1985) 10 figs, 7 refs (in German)

KEY WORDS: Planet gears

A mathematical model and the corresponding computer program for the calculation of dynamic response of planetary gears is presented. The program is illustrated by using it in the analysis of two different gears.

86-1570
Noise Generated by the 1500 and 2000 kW Tapered Spur Gears in the Conveyor Belt System Drives (Geräuschverhalten von 1500 und 2000 kW Kegelstirnrad Getrieben in Antrieben von Förderbandanlagen)
P. Zenker
Konstruktion, 37 (11), pp 423-426 (Nov 1985) 6 figs, 8 refs (in German)

KEY WORDS: Gears, Belt conveyors, Mines (excavations), Noise measurement

The coal from the open pit mining in the Rhine region is transported by means of conveyor belts at about 40,000 t/h. Their drives comprise two step 1500 and 200 kW capacity tapered spur gears At present such gears are regarded as conforming to noise emission standards if they fall in class C of the VDI Directive 2159 (May 1970). The measured confirming data for the majority of the above drives are presented.

COUPLINGS

86-1571
Design and Application Criteria for Connecting Couplings
E.I. Rivin
Wayne State Univ., Detroit, MI
J. Mech., Transm., Autom. in Des., 108 (1), pp 96-105 (Mar 1986) 9 figs, 30 refs

KEY WORDS: Couplings

A classification of couplings as rigid, misalignment-compensating, torsionally flexible, and combination purpose is proposed. Selection criteria for two basic subclasses of misalignment-compensating couplings are derived, some standard designs are analyzed, and modifications of Oldham and gear couplings in which compensatory motion is accommodated by internal shear in thin elastomeric layers instead of reciprocal sliding are described. The new designs demonstrate very high efficiency and exert substantially reduced forces on the connected shafts. Factors determining the influence of torsionally flexible couplings on transmission dynamics are formulated as reduction of torsional stiffness, enhancement of damping, modification of nonlinearity, and inertia distribution. Compensation properties of combination purpose couplings are investigated analytically and a "design index" is introduced. A comparison of important characteristics of some commercially available types of combination purpose couplings is performed, to facilitate an intelligent comparison and selection of various coupling types. A line of approach for the improvement of torsionally flexible/combination purpose couplings by using highly nonlinear elastomeric elements is suggested.

LINKAGES

86-1572
Pumping Action of Aligned Smooth Face Seals Due to Axial Vibrations-Experiment
M. Kaneta, M. Fukahori
Kyushu Institute of Technology, Kitakyushu, Japan
J. Trib., Trans. ASME, 108 (1), pp 46-52 (Jan 1986) 15 figs, 13 refs

KEY WORDS: Seals, Axial vibration, Experimental data

The mechanism of pumping action caused by axially vibrating one of the seal faces with continuous parallel film thickness is clarified experimentally. The results obtained in the experiment confirm the theory developed in a previous investigation. The fundamental design principles of mechanical seals are also described from the viewpoint of this pumping phenomenon.

86-1573
Experimental Stiffness of Tapered Bore Seals
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NASA Lewis Res. Ctr., Cleveland, OH
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 91-94 (Jan 1986) 6 figs, 4 tables, 10 refs

KEY WORDS: Seals, Stiffness coefficients

The stiffness of tapered-bore ring seals was measured with air as the sealed fluid. Static stiffness agreed fairly well with results of a previous analysis. Cross-coupled stiffness due to shaft rotation was much less than predicted. Part of the disparity may be due to simplifying assumptions in the analysis; however, these do not appear to account for the entire difference observed.

STRUCTURAL COMPONENTS

STRINGS AND ROPES

86-1574
Nonlinear Forced Oscillations of a String — 1st Report: Two Types of Responses Near the Second Primary Resonance Point
K. Yasuda, T. Torii
Nagoya Univ., Nagoya, Japan
Bull. JSME, 28 (245), pp 2699-2706 (Nov 1985) 9 figs, 6 refs

KEY WORDS: Strings, Periodic excitation, Subharmonic oscillations, Sum and difference frequencies

Nonlinear dynamic response of a string subjected to periodic excitation will differ qualitatively from the usual harmonic response, because the natural frequencies of a string are in the ratio of integers. To demonstrate that string behavior, a typical case is considered and a theoretical analysis is conducted. The case considered here is the response near the second primary resonance point. The theoretical analysis show that, in addition to the usual harmonic oscillation, two other types of oscillations can occur. One is an oscillation that contains, as its principal components, the subharmonic oscillation of orders 1/2 and 3/2, and the other the so-called summed and differential harmonic oscillation. The conditions under which these types of oscillations occur are obtained, and the characters of the oscillations are discussed. Experimental analysis is also conducted with use of a thin steel strip. The occurrence of the above two types of oscillations is ascertained experimentally, and the validity of the theoretical analysis is confirmed.

CABLES

86-1575
The Effect of Cable Dynamics on the Station-Keeping Response of a Moored Offshore Vessel
K.A. Ansari, N.U. Khan

Gonzaga Univ., Spokane, WA
J. Energy Resources Tech., Trans. ASME, 108 (1), pp 52-58 (Mar 1986) 6 figs, 1 table, 31 refs

KEY WORDS: Moorings, Cables, Ship anchors, Off-shore structures

An anchoring system for an offshore structure must meet certain prescribed requirements controlled by factors such as the site environment, operational constraints and the vessel employed. Its adequacy, survival and ability to stay on site must, therefore, be checked out with proper methods of analysis. The inclusion of cable dynamics is an important consideration in the dynamic analysis of a moored vessel. In this paper, mooring line equations of motion are derived for a multi-component, n-segment model using Lagrange's modified equation, permitting anchor motion, and then numerically solved to yield time histories of cable displacements and cable tensions for the various cable configurations that can occur. Initial conditions can be provided through the mooring line static catenary equations. The nonlinear restoring force terms in the vessel equations of motion are generated through the dynamic tension- displacement characteristics of individual lines. An example involving a moored production barge is examined and results are compared with those of previous work in which a quasi-static cable configuration is employed.

86-1576
Vortex-Induced Vibration and Drag Coefficients of Long Cables Subjected to Sheared Flows
Y.-H. Kim, R. Holler, J.K. Vandiver
Korea Inst. of Technology, Daejeon, Chung Nam-Oo, Korea
J. Energy Resources Tech., Trans. ASME, 108 (1), pp 77-83 (Mar 1986) 10 figs, 2 tables, 11 refs

KEY WORDS: Cables, Fluid-induced excitation, Vortex-induced vibration, Drag coefficients

The vortex-induced vibration response of long cables subjected to vertically sheared flow was investigated in two field experiments. In a typical experiment, a weight was hung over the side of the research vessel by a cable that was instrumented with accelerometers. A typical experiment measured the acceleration response of the cable, the current profile, the tension, and the angle of inclination at the top of the cable. Total drag force was computed from the tension and angle measurement. Two braided Kevlar cables were tested at various lengths from 100 to 9,050 ft. As a result of these experiments, several important conclusions can be drawn. The wave propagation along the length of the cable was damped, and therefore, under most condition the cable behaved like an infinite string. Response spectra were quite broad-band, with center frequencies determined by the flow speed in the region of the accelerometer. Single mode lock-in was not observed for long cables in the sheared current profile. The average drag coefficient of long cables subjected to sheared flow was considerably lower than observed on short cables in uniform flows. The rms response was higher in regions of higher current speed. A new dimensionless parameter is proposed that incorporates the properties of the cable as well as the sheared flow. This parameter is useful in establishing the likelihood that lock-in may occur, as well as in estimating the number of modes likely to respond.

86-1577
Sea Bed-Structure Interaction for Submarine Cables
C.Z. Karakostas, C.C. Baniotopoulos, P.D. Panagiotopoulos
Aristotle Univ., Thessaloniki, Greece
Computers Struc., 22 (3), pp 213-224 (1986) 9 figs, 7 tables, 16 refs

KEY WORDS: Cables, Submarines

In this study, the dynamic behavior of a submarine cable constrained by a frictionless rigid sea profile is studied. After a suitable local and time discretization, the arising unilateral contact-impact problem is formulated as a sequence of variational inequality problems which lead equivalently to a sequence of quadratic programming problems. The solution method is based on some propositions which constitute the mechanical interpretation of some theorems of quadratic programming and combines the advantages of the optimization algorithms with advantages of the trial and error methods. Finally the applicability of the proposed method is illustrated by means of a numerical example.

86-1578
On the Dynamics of a Multicomponent Mooring Line
N.U. Khan, K.A. Ansari
Univ. of Petroleum and Minerals, Dhahran, Saudi Arabia
Computers Struc., 22 (3), pp 311-334 (1986) 12 figs, 17 refs

KEY WORDS: Moorings, Cables, Lagrange equations

The inclusion of cable dynamics is an important consideration in the evaluation of the dynamic station-keeping response of a moored offshore vessel. In this paper, equations of motion for a multicomponent mooring cable are derived allowing for anchor motion, if any, and then numerically solved to yield time histories for the various cable configurations that can occur. From the time histories obtained, the nonlinear dynamic tension-displacement characteristics of the mooring line can be derived which would be useful in providing the restoring force terms occurring in the equations of motion of a moored offshore vessel. An example involving a typical anchoring line used on a moored production barge is examined to demonstrate the applicability of the analysis presented.

BARS AND RODS

86-1579

Elastodynamic Response of a Strip with an Edge Crack to the Action of Antiplane Shear Sudden Loading

Li Kerong, Tai Weihua
Huazhong Univ. of Science and Technology
Acta Mech. Solida Sinica, 3, pp 317-326 (1985) 5 figs, 11 refs

KEY WORDS: Strips, Cracked media, Elastodynamic response

The problem of transient response of a strip with an edge crack to the action of an antiplane shear sudden loading on the face of crack is investigated. There are two cases to be considered. Both edges of the strip are free. One of the edges is free and the other is clamped. The Laplace and Fourier transforms are used. The problem is reduced to solving a Cauchy singular integral equation in Laplace domain. The numerical results of dynamic stress intensity factors and relative displacements of crack surface are given.

86-1580

Simple Model for Reinforcing Bar Anchorages Under Cyclic Excitations

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California Univ., Berkeley, Earthquake Engineering Res. Ctr.
Rept. No. UCB/EERC-85/05, 77 pp (Mar 1985)
PB86-112919/GAR

KEY WORDS: Bars, Hysteretic damping, Cyclic loading

A simple and computationally attractive model of the hysteretic response of reinforcing bar anchorages under severe cyclic excitations is presented The model is based on a piecewise linear approximation of the bond stress distribution function along the bar anchorage length. Only three intermediate points located between the ends of the bar suffice to yield results which match the finite element solution of the problem. The bond stress value at these intermediate points is established by iteratively satisfying the equilibrium and compatibility equations of the bond problem. The proposed model is used to calculate the response of the bottom reinforcing bars of an interior beam-column joint subjected to deformation reversals of increasing magnitude, as well as the response of a single reinforcing bar subjected to cyclic push-pull. The results are compared with the finite element solution of the studied cases.

86-1581

Errors in Response Calculations for Extensional Vibrations of Bars

H. Wada
Tohoku Univ., Sendai, Japan
Earthquake Engrg. Struc. Dynam., 14 (1), pp 121-132 (Jan-Feb 1986) 4 figs, 7 tables, 9 refs

KEY WORDS: Bars, Finite element technique, Laplace transformation, Normal mode method, Newmark method

The transient extensional vibrations of a slender and uniform bar, which is clamped at one end and is subjected to an axial force at the other free end, are investigated. Three methods: the Laplace transform method, the normal mode and Newmark beta methods in conjunction with the finite element method (FEM), are applied. The errors caused by the spatial discretization of the FEM and the direct integration of the Newmark beta method are studied and compared with those of the previous paper where the flexural vibration of a cantilever beam were considered. The reason why the extentional vibration problem is investigated here is that the condition seems to be severe due to the closeness of adjacent natural frequencies. The numerical results show that the errors in response of the extensional vibration problem are large. However, if one follows the criterion proposed in this paper, accurate response is obtainable by the Newmark beta method, which requires less computer time than any of the other methods mentioned above.

86-1582
Effects of Torsional Moment on the Transverse Vibrations of Circular Bars
A. Hagita, T. Kuno, M. Mizuno
Mitsubishi Heavy Industries, Kanagawa-ken, Japan
Bull. JSME, 29 (247), pp 187-193 (Jan 1986) 4 figs, 7 refs

KEY WORDS: Circular bars, Flexural vibrations, Torque

IN this paper, the effects of the torsional moment on the free transverse vibration of circular bars are investigated with the condition that the axial elongation can be neglected. A differential equation on the non-planar motion of the bar is derived. Then the equation is solved by means of the perturbation method and the numerical one for the case in which both ends are clamped. The results show that the characteristic frequency decreases quadratically with an increase of the torsional moment, which is contrary to the case of tension. It is also confirmed that the non-planar vibration arises due to the torsional moment.

BEAMS

86-1583
A Finite Difference Method for the Free Vibration Analysis of Stepped Timoshenko Beams and Shafts
A.S. Sarigul (Aydin), G. Aksu
Middle East Technical Univ., Gaziantep, Turkey
Mech. Mach. Theory, 21 (1), pp 1-12 (1986) 10 figs, 4 tables, 11 refs

KEY WORDS: Beams, Shafts, Variable cross section, Timoshenko theories, Finite difference technique

A method based on the variational principles in conjunction with the finite difference technique is used to examine the free vibrational characteristics of Timoshenko beams and shafts. The interlacing grid technique is used to express the strain energy of modal subdomains and the partial derivatives appearing in the functionals are replaced by the finite difference equations in terms of discrete displacement and rotational components. The developed technique is applied to dynamic analysis of uniform and nonuniform stepped thickness beams and shafts.

86-1584
Effective Length Factor for Restrained Beam-Column
Kuo-Kuang Hu, D.C. Lai
ASCE J. Struc. Engrg., 112 (2), pp 241-256 (Feb 1986) 13 figs, 1 table, 7 refs

KEY WORDS: Beam-columns, Off-shore structures, Elastic restraints, Critical loads

A development was undertaken to demonstrate the feasibility of using computer methods to calculate the effective length factors of elastically restrained beam-columns. Three theoretical models for elastically restrained beam-columns were derived. These beam-columns are restrained by elastic rotational springs at the ends and an elastic translational spring at any given location along the beam-columns.

86-1585
Stability of a Rotating Pretwisted Non-Uniform Cantilever with a Tip Mass Subjected to Dissipative and Follower Forces
R.C. Kar
Indian Institute of Technology, Kharagpur, India
Computers Struc., 22 (2), pp 115-122 (1986) 6 figs, 13 refs

KEY WORDS: Cantilever beams, Mass-beam systems, Follower forces, Dynamic stability

The stability of a rotating pretwisted non-uniform viscoelastic cantilever beam subjected to a tangential force applied at its free end is determined by a method of approximation based on an adjoint variational principle. The coupled equations of motion are derived from a conservation law, the adjoint boundary value problem is introduced, and an approximate stability determinant is developed from the variational principle. The stability determinant is solved numerically for a variety of choices of values for the rotary inertia of the beam, transverse and rotary inertia properties of a mass capping the free end, the rotational speed, and the pretwist angle, and several graphs are presented to show the influence of these parameters upon the value of the critical flutter load.

86-1586
On the Transverse Vibration of Beams of Rectangular Cross-Section
J.R. Hutchinson, S.D. Zillmer
Univ. of California, Davis, CA
J. Appl. Mech., Trans. ASME, 53 (1), pp 39-44 (Mar 1986) 11 figs, 10 refs

KEY WORDS: Rectangular beams, Natural frequencies, Flexural vibrations, Timoshenko theory

An exact solution for the natural frequencies of transverse vibration of free beams with rectangular cross-section is used as a basis of comparison for the Timoshenko beam theory and a plane stress approximation which is developed herein. The comparisons clearly show the range of applicability of the approximate solutions as well as their accuracy. The choice of a best shear coefficient for use in the Timoshenko beam theory is considered by evaluation of the shear coefficient that would make the Timoshenko beam theory match the exact solution and the plane stress solution. The plane stress solution is shown to provide excellent accuracy within its range of applicability.

86-1587
Free Vibration of Statically Compressed Clamped beams on Nonlinear Elastic Foundation
V. Birman
Univ. of New Orleans, New Orleans, LA
Mech. Res. Comm., 12 (5), pp 303-308 (Sept/Oct 1985) 3 figs, 7 refs

KEY WORDS: Beams, Elastic foundations, Natural frequencies

In this paper a free nonlinear vibration of a clamped beam undergoing axial loading and resting on the nonlinear (cubic) elastic foundation is considered. The amplitudes of vibration are assumed to remain in the range where the single mode analysis is valid. The effect of an axial compressive force on the frequencies and the influence of the elastic foundation are shown in numerical examples.

CYLINDERS

86-1588
Vortex-Induced Response of a Flexible Cylinder in a Sheared Current
N.M. Patrikalakis, C. Chryssostomidis
Massachusetts Institute of Technology, Cambridge, MA
J. Energy Resources Tech., Trans. ASME, 108 (1), pp 59-64 (Mar 1986) 7 figs, 3 tables, 18 refs

KEY WORDS: Cylinders, Fluid-induced excitation, Vortex shedding, Marine risers

In this paper, a method for the approximate prediction of the static and lift responses of a flexible cylinder in a unidirectional variable stream is outlined. This prediction is based on information derived from experimental results involving rigid cylinders forced to oscillate sinusoidally orthoganally to a uniform stream. This approach represents the multifrequency lift response of a flexible cylinder in a sheared current by predicting a number of independently determined, monochromatic, multimode dynamic solutions. For each such multimode solution, this procedure allows the simultaneous evaluation of lift response frequency, amplitude, and phase between modes. A numerical example assuming bimodal solutions is included to illustrate this method for the geometry of a single-tube marine riser.

86-1589
Free Vibration of Axisymmetrical Solid Bodies with Meridionally Varying Profile
T. Irie, G. Yamada, I. Okada
Hokkaido Univ., Sapporo, Japan
J. Acoust. Soc. Amer., 79 (2), pp 375-381 (Feb 1986) 6 figs, 3 tables, 9 refs

KEY WORDS: Cylinders, Bodies of revolution, Natural frequencies, Mode shapes, Ritz method

An analysis is presented for the three-dimensional vibration problem of determining the natural frequencies and the mode shapes of axisymmetrical solid bodies, with meridionally varying profiles, expressed as an arbitrary function. For this purpose, the body is transformed into a circular cylinder with unit axial length and unit radius, by a transformation of variables. With the displacements of the transformed cylinder assumed in the forms of algebraic polynomials, the dynamical energies of the cylinder are evaluated, and the frequency equation is derived by the Ritz method. This method is applied to barrel or hourglass-type bodies and frustums of cone, under two combinations of boundary conditions at the ends, and the natural frequencies and the mode shapes are calculated, numerically giving the results.

FRAMES AND ARCHES

86-1590
Static and Dynamic Behavior of Monotube Span-Type Sign Structures. Volume 1
M.R. Ehsani
Arizona Transportation and Traffic Inst., Tucson, AZ
Rept. No. FHWA/AZ-84/194-1, 102 pp (May 1984) PB86-110798/GAR

KEY WORDS: Traffic sign structures, Finite element technique

The report presents the results of the first major investigation into the static and dynamic behavior characteristics of monotube span-type sign structures. Detailed static and dynamic stresses and deflections have been determined for an actual 100 ft. span sign structure, utilizing two- and three-dimensional finite element modeling. Parametric studies have also been made, where the effects of column stiffness, beam stiffness, span, and sign location and size were examined. It is shown that in-plane and out-of-plane analyses can be conducted independently, and that stresses for tubular members can be determined by vector addition. Design recommendations are made on the basis of stress and deflection computations for simple planar frames. Cambering is recommended for structures where gravity load deflections may be aesthetically undesirable.

86-1591
Static and Dynamic Behavior of Monotube Span-Type Sign Structures. Volume 2
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Arizona Transportation and Traffic Inst., Tucson, AZ
Rept. No. FHWA/AZ-84/194-11, 78 pp (May 1984) PB86-111721/GAR

KEY WORDS: Traffic sign structures, Finite element technique

The report is the second volume of a two volume work on the results of investigations of static and dynamic behavior characteristics of monotube span-type sign structures.

PANELS

86-1592
Optimal Forms of Shallow Cylindrical Panels with Respect to Vibration and Stability
R.H. Plaut, L.W. Johnson
J. Appl. Mech., Trans. ASME, 53 (1), pp 135-140 (Mar 1986) 5 figs, 2 tables, 24 refs

KEY WORDS: Panels, Optimization, Geometric effects, Fundamental frequencies, Buckling

Thin, shallow, elastic, cylindrical panes with rectangular planform are considered. The midsurface form which maximizes the fundamental frequency of vibration, and the form which maximizes the buckling value of a uniform axial load is sought. The material, surface area, and uniform thickness of the panel are specified. The curved edges are simply supported, while the straight edges are either simply supported or clamped. For the clamped case, the optimal panels have zero slope at the edges. In the examples, the maximum fundamental frequency is up to 12 percent higher than that of the corresponding circular cylindrical panel, while the buckling load is increased by as much as 95 percent. Most of the solutions are bimodal, while the rest are either unimodal or trimodal.

PLATES

86-1593
The Nonlinear Dynamic Response of an Elastic-Plastic Thin Plate Under Impulsive Loading
Wang, Xintian
Acta Mech. Solida Sinica, 3, pp 281-295 (1985) 16 figs, 2 tables, 18 refs CSTA No. 531-85.80

KEY WORDS: Plates, Elastic plastic properties, Impact excitation, Nonlinear response

In this paper the effects of the physical and geometrical nonlinearities in a thin plate are treated as equivalent body forces and equivalent loads. Using the concept of influence functions, we present a method for analyzing the thin plate problem with both kinds of nonlinear effects In the calculation of practical examples the numerical solutions for nonlinear dynamic responses of an elastic-plastic thin plate are obtained for various hardening coefficients and different impact loads, all of the results are quite regular.

86-1594
Ultimate Load Behavior of Reinforced Concrete Plates and Shells Under Dynamic Transient Loading
G.Q. Liu, D.R.J. Owen
Tong-Ji Univ.

Intl. J. Numerical Methods Engrg., 22 (1), pp 189-208 (Jan 1986) 16 figs, 32 refs

KEY WORDS: Plates, Shells, Reinforced concrete, Transient excitation, Finite element technique

This paper considers a layered thick shell finite element procedure for determining the dynamic transient nonlinear response of plates and shells. The degenerated three-dimensional isoparametric shell element with independent rotational and translational degrees-of-freedom is employed. A layered formulation is adopted to represent the steel reinforcement and to simulate progressive concrete cracking through the thickness. The dynamic yielding function is assumed to be a function of the current strain rate, in addition to being total plastic strain or work dependent. The concrete model also simulates both compressive crushing and tensile cracking behaviors and in implicit Newmark algorithm is employed for time integration of the equations of motion. Several numerical examples are presented for both slab and shell structures and the results obtained compared with those from other sources wherever available.

86-1595
Vibrations of Thin Elastic Plates with Point Supports: A Comparative Study
J.C. Utjes, P.A.A. Laura, G. Sanchez Sarmiento, R. Gelos
Empresa Nuclear Argentina de Centrales Electricas SA, Buenos Aires, Argentina
Appl. Acoust., 19 (1), pp 17-24 (1986) 2 figs, 4 tables, 9 refs

KEY WORDS: Plates, Supports, Natural frequencies, Finite element technique, Ritz method

This paper deals with vibrating plates of different shapes and different support arrangements. Natural frequencies are determined using a finite element algorithm and compared with results available in the literature where possible. It is also shown that in the case of clamped and simply supported plates of regular polygonal shape with a central clamping support simple polynomial expressions yield accurate values for the fundamental frequency coefficient when the Ritz method is used.

86-1596
Experimental Determination of the Damping Ratio of a Thermal Insulation Cover Plate
H. Fenech
Univ. of California, Santa Barbara, CA
Appl. Acoust., 19 (1), pp 47-60 (1986) 5 figs, 3 refs

KEY WORDS: Rectangular plates, Damping coefficients

This paper describes an experiment on the measurement of the damping ratio of a square plate cover with kaowool fiber insulation backup. The measured values of the camping ratio are given for two different sizes of plate with different attachments. The effects of the configuration and attachments on the damping ratio are discussed.

86-1597
Step-Load Response Analysis by Generalized Functions
E. Kessler
ASCE J. Engrg. Mech., 112 (3), pp 311-321 (Mar 1986) 3 figs, 8 refs

KEY WORDS: Rectangular plates, Step functions, Heaviside functions, Blast resistant structures

Blast loads on structures are conveniently represented by means of Heaviside step functions. Due to the nondifferentiability of these functions the analytical handling (integration by parts) fails quickly. This problem can be successfully overcome by applying the theory of generalized functions. Calculations for rectangular plates are presented for illustration.

86-1598
On the Effect of Free, Rectangular Cut-Outs Along the Edge on the Transverse Vibrations of Rectangular Plates
P.A.A. Laura, J.C. Utjes, V.H. Palluzzi
Institute of Applied Mechanics, Puerto Belgrano Naval Base, Argentina
Appl. Acoust., 19 (2), pp 139-151 (1986) 6 figs, 2 tables, 11 refs

KEY WORDS: Rectangular plates, Flexural vibrations, Rayleigh-Ritz method, Finite element technique, Hole containing media

The title problem is tackled using two alternative approaches: a simple Rayleigh-Ritz methodology and a finite element algorithmic procedure. Numerical results are obtained in the case of simply supported and clamped plates with free, square holes located at the middle of a plate edge. Experimental values are obtained in the case of clamped plates. It is concluded that the agreement is very good from an engineering viewpoint as long as the hole size is moderate. Since this constitutes a valid restriction in practice when ducts, pipes, etc., traverse plates or slaps, one can say that the Rayleigh-Ritz procedure yields fundamental frequency coefficients which possess sufficient accuracy from a designer's viewpoint.

86-1599
Nonlinear Dynamic Response of Rectangular Plates
S.A. Ali, S.I. Al-Noury
Washington Univ., St. Louis, MO
Computers Struc., 22 (3), pp 433-437 (1986) 5 figs, 11 refs

KEY WORDS: Rectangular plates, Nonlinear response, Finite difference technique

The coupled differential equations of motion of rectangular plates are developed using nonlinear strain-displacement relationships. The equations of motion are solved by an implicit finite difference method applied to a plate subjected to a sudden uniform pressure and with simply supported and clamped boundary conditions. Examples involving large amplitude vibrations of plates are presented.

86-1600
Joint Research Effort on Vibrations of Twisted Plates, Phase 1: Final Results
R.E. Kielb, A.W. Leissa, J.C. MacBain, K.S. Carney
NASA, Lewis Res. Ctr., Cleveland, OH
NASA-RP-1150, 100 pp (Sept 1985) N86-10579/8/GAR

KEY WORDS: Cantilever plates, Turbine blades

The complete theoretical and experimental results of the first phase of a research study on the vibration characteristics of twisted cantilever plates are given. The study is conducted to generate an experimental data base and to compare many different theoretical methods with each other and with the experimental results. Plates with aspect ratios, thickness ratios, and twist angles representative of current gas turbine engine blading are investigated. The theoretical results are generated by numerous finite element, shell, and beam analysis methods. The experimental results are obtained by precision matching a set of twisted plates and testing them at two laboratories. The second and final phase of the study will concern the effects of motion.

SHELLS

86-1601
Acoustical Behavior of a Cylindrical Thin Shell at Low Frequencies (Comportement acoustique d'un tube cylindrique mince en basse frequence)
J.L. Rousselot
L.C.T., Velizy-Villacoublay, France
Acustica, 58 (5), pp 291-297 (Oct 1985) 8 figs, 11 refs (in French)

KEY WORDS: Shells, Wave propagation, Sound waves

Theoretical and experimental investigation of circumferential waves propagating along the surface of convex elastic objects have been of interest for several years. Two ways have been employed to describe these waves circulating on spherical and cylindrical surfaces: Modal theory and Sommerfeld-Watson integral transformation. Results of these methods agree very well for massive cylinder and tube when the wall thickness is large enough. For very thin walls, the relation is not so easy to establish and apparent anomalies are investigated in the present paper. This study shows that at very low frequencies Rayleigh waves split into two waves. Curves of dispersion and of attenuation coefficient versus frequency are presented. The effects result from re-emission of energy into the ambient fluid. Results are compared with recently published papers. The characteristic of one of the waves can be seen as a Stoneley at low frequency and Franz wave at higher frequencies. Some pressure diagrams illustrate the re-radiating properties of this kind of objects.

86-1602
Forced Vibrations of Elastic Shallow Shell Due to the Moving Mass
Cheng Xiang-sheng
Appl. Math. Mech., 6 (3), pp 233-240 (1985) CSTA No. 519-85.78

KEY WORDS: Shells, Moving loads, Resonant response, Critical speeds

This paper discussed the forced vibrations of the elastic shallow shell due to the moving mass by means of the variational method. A series of problems such as the forced vibrations, resonance conditions and critical speeds are considered.

86-1603
Axisymmetric Static and Dynamic Buckling of Orthotropic Truncated Shallow Conical Caps
P.C. Dumir, K.N. Khatri
Indian Institute of Technology, New Delhi, India
Computers Struc., 22 (3), pp 335-342 (1986) 12 figs, 1 table, 12 refs

KEY WORDS: Caps, Dynamic buckling

This investigation deals with the axisymmetric static and dynamic buckling of a cylindrically orthotropic truncated shallow conical cap with clamped edge. The cases of conical caps with a free central circular hole and with a hole plugged by a rigid central mass shave been considered. Dynamic load is taken as a step function load. The influence of orthotropic parameter and annular ratio on the buckling loads has been investigated. New results for static and dynamic buckling loads have been presented for the isotropic and orthotropic truncated conical caps. Dynamic buckling loads obtained form static analysis have been found to agree well with the dynamic buckling loads based on transient response.

86-1604
Nonlinear Axisymmetric Response of Orthotropic Thin Spherical Caps on Elastic Foundations
P.C. Dumir
Indian Institute of Technology, New Delhi, India
Intl. J. Mech. Sci., 27 (11/12), pp 751-760 (1985) 7 figs, 3 tables, 16 refs

KEY WORDS: Caps, Spherical shells, Elastic foundations

This paper presents an approximate analytical solution of the large deflection axisymmetric response of polar orthotropic thin spherical caps resting on elastic foundations. The Winkler, nonlinear Winkler and Pasternak models of the foundations are considered. Caps with elastically restrained edges are analyzed. Donnell type equations are employed. One term approximation is made for the deflection and the Galerkin's method is used to get the governing equation for the apex deflection. Nonlinear free vibration response, static response under uniformly distributed load, and the maximum transient response under uniformly distributed step load have been obtained. The effect of the various parameters is investigated.

86-1605
Shallow Spherical Shells on Pasternak Foundation
D.N. Paliwal, S.N. Sinha, B.K. Choudhary, Jr.
Motilal Nehru Regional Engrg. College, Uttar Predesh, India
ASCE J. Engrg. Mech., 112 (2), pp 175-182 (Feb 1986) 3 figs, 10 refs

KEY WORDS: Spherical shells, Pasternak foundations, Berger theory, Natural frequencies

Static and dynamic analysis of fully clamped shallow spherical shells, subjected to uniform normal pressure on concave side is made using Berger's and modified Berger's techniques. They are supported on a Pasternak foundation. Expressions for large static deflections are obtained and results compared with those of Nash and Modeer. The preceding techniques are employed to get the approximate expressions for natural frequency. Values of nondimensional natural frequency obtained by various methods are compared with results of Reissner, Nowacki, Bucco and others. While a modified Berger's approach gives very good results for static analysis of fully clamped shells, the original Berger's approach is found to be more suitable for free vibration analysis, as results tally with those obtained by Reissner, using Rayleigh's method, even for higher ratios of (H/h). The modified Berger's technique gives sufficiently accurate values of natural frequencies only for smaller values of (H/h).

86-1606
On the Wave Propagation in an Elastic Hollow Cylinder with Long-Range Cohesion Forces
J.L. Nowinski
Univ. of Delaware, Newark, DE
J. Appl. Mech., Trans. ASME, 53 (1), pp 121-124 (Mar 1986) 1 fig, 25 refs

KEY WORDS: Cylindrical shells, Wave propagation

After a brief derivation of the formula for the nonlocal moduli, Fourier transforms of the stress components in their nonlocal aspect are established. Satisfaction of the traction-free boundary conditions leads to the frequency equation of the problem. A particular case involving longitudinal Lame modes is analyzed in more detail. A numerical example solved shows a considerable decrease of the speed and the frequency of the short waves as compared with those of long waves studied in the classical theory.

86-1607
Correlation between Vibrations and Buckling of Cylindrical Shells Stiffened by Spot-Welded and Riveted Stringers
T. Weller, H. Abramovich, J. Singer
Technion - Israel Inst. of Tech., Haifa, Israel
Rept. No. TAE-537, 81 pp (Jan 1985) N86-10584/8/GAR

KEY WORDS: Cylindrical shells, Rivets/joints, Welded joints, Nondestructive tests, Vibratory techniques

The Vibration Correlation Techniques (VCT) was successfully applied, as a nondestructive tool for definition of actual boundary conditions and better prediction of buckling loads, to shells with spot-welded and riveted stringer stiffeners. It was demonstrated that the effective boundary conditions of the shells are fairly well by this technique. The lumping effects of the initial imperfection in the definition of the equivalent restraints are relatively small. Measurements of initial geometrical imperfections were conducted prior to testing of each shell. It is shown that calculation of their degrading effect on the buckling load serves as a complementary tool for improved predictions of actual buckling loads. When combined with the more precisely defined boundary conditions, yielded by the VCT, the agreement between predictions and tests is significantly improved.

PIPES AND TUBES

86-1608
The Stability of Oscillatory Hagen-Poiseuille Flow
J.T. Tozzi, C.H. von Kerczek
USCG Headquarters
J. Appl. Mech., Trans. ASME, 53 (1), pp 187-192 (Mar 1986) 6 figs, 1 table, 22 refs

KEY WORDS: Tubes, Fluid-induced excitation, Periodic excitation

The linear stability theory of the nonzero mean, sinusoidally oscillating flow in a tube of circular cross section is examined. It is found that the relevant axisymmetric disturbances in the oscillatory flow are more stable than the axisymmetric disturbances of the mean flow alone. This result holds for values of the cross-sectional average oscillation velocity amplitude at least as large as seven-tenths the average mean-flow velocity amplitude. Although the instantaneous velocity profile contains generalized inflection rings for a substantial portion of the oscillation period, the disturbances do not become instantaneously unstable at any time, even for very low frequency oscillation.

86-1609
Dynamic Analysis of Fluid-Filled Piping Systems Using Finite Element Techniques
G.C. Everstine
David Taylor Naval Ship Res. and Dev. Ctr., Bethesda, MD
J. Pressure Vessel Tech., Trans. ASME, 108 (1), pp 57-61 (Feb 1986) 6 figs, 2 tables, 26 refs

KEY WORDS: Pipelines, Fluid-filled containers, Finite element technique

Two finite element procedures are described for predicting the dynamic response of general three-dimensional fluid-filled elastic piping systems. The first approach, a low frequency procedure, models each straight pipe or elbow as a sequence of beams. The contained fluid is modeled as a separate coincident sequence of axial members (rods) which are tied to the pipe in the lateral direction. The model includes the pipe hoop strain correction to the fluid sound speed and the flexibility factor correction to the elbow flexibility. The second modeling approach follows generally the original Zienkiewicz-Newton scheme for coupled fluid-structure problems except that the velocity potential is used as the fundamental fluid unknown to symmetrize the coefficient matrices. From comparisons of the beam model predictions to both experimental data and the three-dimensional model, the beam model is validated for frequencies up to about two-thirds of the lowest fluid-filled lobar pipe mode. Accurate elbow flexibility factors are seen to be important for effective beam modeling of piping systems.

86-1610
Simulation of Fluid Network Dynamics by Transmission Line Modelling
R.F. Boucher, E.E. Kitsios
Univ. of Sheffield, England
IMechE. Proc., Part C: Mech. Engrg. Sci., 200 (C1), pp 21-29 (1986) 9 figs, 11 refs

KEY WORDS: Pipelines, Simulation, Transmission lines

The classical solution to the acoustic wave equation reveals decoupled left-and right-going waves carrying pressure and flow. Using a variant of these waves representing power, it is shown that such decoupled waves permit fluid transmission lines in circuits to be modeled simply as pure time delays. It is also demonstrated that lumped inertance and capacitance may be modeled as transmission line stubs. Thus all dynamic elements in a circuit are represented as pure time delays and the only computations required are of wave scattering at junctions. Linear resistance is most easily admitted, especially if it is concentrated at the junctions. Computations on simple circuits are presented and shown to compare favorably with classical lumped analyses. The significance of various approximations made is discussed. The method may be applied to fluid circuits of arbitrary size and complexity, and to circuits of other wave-propagating elements.

86-1611
Flow-Induced Vibration Caused by Roughness in Pipes Conveying Fluid
A. Shulemovich
J. Appl. Mech., Trans. ASME, 53 (1), pp 181-186 (Mar 1986) 5 figs, 28 refs

KEY WORDS: Pipes, Fluid-induced excitation, Surface roughness

This paper presents a theoretical investigation of self-excited vibrations of pipes conveying fluid due to roughness. A model of a laminar friction, considered as the excitation mechanism, is based on Prandtl's universal velocity distribution for the turbulent boundary layer and on Nikuradse's experiments. The analysis has shown that the friction characteristic has a negative slope in a certain range of fluid velocities. This range is a function of pipe roughness and is shifted to lower flow velocities due to roughness growth during pipe operation. It was found that the differential operator of a piping loop motion based on the nonlinear restoring characteristic coincides with the differential operator of Duffing's equation for the hardening system. The energy method was used to obtain the approximate closed-form solution for the amplitude of steady self-excited vibrations. The unstable response with jump phenomena can appear due to interaction of small turbulent disturbances in conveying fluid with a given nonlinear system.

DUCTS

86-1612
Dispersion Relation of Pressure Waves in a Swirling Flow (Relation de dispersion des ondes de pression dans un écoulement tournant)
M. Roger, H. Arbey
Ecole Centrale de Lyon, Ecully, France
Acustica, 59 (2), pp 95-101 (Dec 1985) 7 figs, 14 refs (in French)

KEY WORDS: Ducts, Wave propagation, Modal analysis

This paper is devoted to a modal theory of the propagation of pressure waves in a duct with axisymmetric swirling flow. By virtue of the mean rotation, a coupling between fluctuations of vortical and acoustic types is pointed out in the vorticity equation and in the pressure equation. This occurs even in the linear theory presented. The dispersion relation of the duct is given for spinning modes, showing that pressure waves propagate at once as inertial modes and acoustic modes. First are then induced by complementary accelerations. The cut-off frequencies of spinning acoustic modes are systematically modified by the mean rotation.

86-1613
Oscillations of Subsonic Flow in an Abruptly Expanding Circular Duct
W.C. Selerowicz, A.P. Szumowski, G.E.A. Meier
Max-Planck-Inst. fuer Stroemungsforschung, Goettingen, Fed. Rep. Germany
Rept. No. MPIS-16/1984, 39 pp (Sep 1984) N86-10028/6/GAR

KEY WORDS: Ducts, Fluid-induced excitation

Subsonic flow in an abruptly expanding duct driven from a pressure reservoir, which shows self-excited oscillations of considerable amplitude was studied by visualization and acoustic pressure measurements. Depending on the pressure drop in the duct and its dimensions, three distinct types of flow oscillations are observed. These are characterized by symmetrical and antisymmetrical oscillations of the separated jet in the abruptly expanding section, and symmetrical oscillations of the jet periodically separating and reattaching to the wall. The oscillations are accompanied by intensive noise, with an intensity of 40 dB above the turbulence noise level.

86-1614
Influence of Errors on the Two-microphone Method for Measuring Acoustic Properties in Ducts
H. Bodén, M. Åbom
Royal Inst. of Technology, Stockholm, Sweden
J. Acoust. Soc. Amer., 79 (2), pp 541-549 (Feb 1986) 7 figs, 12 refs

KEY WORDS: Ducts, Acoustic properties, Two microphone technique, Measurement techniques, Error analysis

Using the two-microphone method, acoustic properties in ducts, as, for example, reflection coefficient and acoustic impedance, can be calculated from a transfer function measurement between two microphones. In this paper, a systematic investigation of the various measurement errors that can occur and their effect on the calculated quantities is made. The input data for the calculations are the measured transfer function, the microphone separation, and the distance between one microphone and the sample. First, errors in the estimate of the transfer function are treated. Conclusions concerning the most favorable measurement configuration to avoid these errors are drawn. Next, the length measurement errors are treated. Measurements were made to study the question of microphone interference. The influence of errors on the calculated quantities has been investigated by numerical simulation. From this, conclusions are drawn on the useful frequency range for a given microphone separation and on the magnitude of errors to expect for different cases.

BUILDING COMPONENTS

86-1615

Application of Folded Plate Analysis to Bending, Buckling and Vibration of Multilayer Orthotropic Sandwich Beams and Panels

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Admiralty Research Establishment, Dunfermline, UK

Computers Struc., 22 (3), pp 491-497 (1986) 5 figs, 9 refs

KEY WORDS: Plates, Beams, Panels, Sandwich structures

A unified analysis method based on two-dimensional elasticity theory is outlined for evaluation of bending, buckling and vibration of multilayer orthotropic sandwich beams and panels. The effects of initial geometric imperfections are included. It is shown that beams or panels deforming under conditions of plain stress or plane strain may be treated as special instances of folded-plate structures using computer programs which are now widely available. Examples are given, including evaluation of stress contours in a sandwich panel under patch load and analysis of overall and local (face-wrinkling) buckling modes in sandwich panels with stiff and soft cores.

86-1616

Audio Eigen-Frequencies of Floors with Edges Elastically Restrained Against Rotation

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Appl. Acoust., 19 (3), pp 203-224 (1986) 2 figs, 5 tables, 14 refs

KEY WORDS: Floors, Natural frequencies

This paper deals with a method for evaluating eigen-frequencies within the acoustic domain and is applied for the analysis of partially fixed plates. For this method the solution assumption relies on an extended form of Kantorovich-Krylov polynomial series. The relations between the coefficients of the series were algebraically defined, taking into account the plate geometry, which results in an equation set of the plate's behavior. Results were compared, when possible, with values available in the literature. Tables of eigenvalues of typical building plates are given for various dimension ratios and strengths of fixing.

DYNAMIC ENVIRONMENT

ACOUSTIC EXCITATION

86-1617

Noise Reduction of Axial-Flow Machines by Means of Wide-Band Acoustic Absorption Near the Grating—Possibilities and Limitations (Lärmminderung bei axialen Strömungsmaschinen durch breitbandige Schallabsorption in Gitternähe—Möglichkeiten und Grenzen)

S. Gruhl, K. Biehn

Zentralinstitute für Arbeitsschutz Dresden, Fed. Rep. Germany

Maschinenbautechnik, 34 (11), pp 502-506 (1985) 7 figs, 1 table, 22 refs (in German)

KEY WORDS: Acoustic absorption, Noise reduction, Acoustic linings

Studies of the efficiency of source lined housings of axial-flow fans show that such housings behave in the acoustic sense as classic absorbtion silencer. The arrangement of absorber near the grating doesn't cause additional damping effects. Because absorbing housings cause considerable decrease of efficiency, this measure of noise reduction doesn't represent an alternative of the customary principle.

86-1618
Description of the Actual Noise of a Two-Stroke Engine by Means of a Linear Mathematical Model (Beschreibung des realen Luftschallverhaltens eines Zweitaktmotors durch ein lineares Ersatzmodell)
N. Kania, J. Schwarz
Universität Hannover, Hannover, Fed. Rep. Germany
Appl. Acoust., 19 (3), pp 225-241 (1986) 10 figs, 15 refs (in German)

KEY WORDS: Engine noise

A linear model is applied to a small, two-stroke crankcase-scavenged engine (swept volume, 70 cm^3; output 3.0 kW at 7500 litres/min) in an attempt to identify the real behavior of airborne noise. This paper deals only with the gas exchange noises generated at the intake and outlet ports which over-ride the influence of the structure-borne noise under the selected operating conditions.

86-1619
In-plane Contribution to Structural Noise Transmission
R.H. Lyon
Massachusetts Institute of Technology, Cambridge, MA
Noise Control Engrg. J. 26 (1), pp 22-27 (Jan-Feb 1986) 12 figs, 1 table, 8 refs

KEY WORDS: Structure borne noise, Shear waves, Longitudinal waves, Multibody systems

Noise transmitted in structures, or structureborne sound, is ordinarily assumed to be dominated by the flexural modes of the structure because these modes couple best with the sound field, are directly excited by impact forces. This view is generally correct for the directly-excited structure, and is probably correct for structures with a few components. However, when the transmission takes place through several elements, then one should consider the contribution to the noise transmission by in-plane longitudinal and shear modes. This paper shows that such transmission may be dominant for multi-component structures at frequencies high enough so that these in-plane modes are present, and that significant error in noise transmission can be made if they are neglected.

86-1620
Inter-Noise 85
Proc. Intl. Conf. Noise Control Engrg., Munich, Fed. Rep. Germany, Dec 8-10, 1985. 2 vols, Spons. I/INCE and Fed. Inst. of Occupational Safety, Fed. Rep. Germany
Avail: Noise Control Foundation, P.O. Box 3469, Arlington Branch, Fed. Rep. Germany, $80.00

KEY WORDS: Noise reduction, Proceedings, Acoustic intensity method, Traffic noise, Machinery noise

There were 351 technical papers presented at this conference covering all areas of noise control engineering. Papers on aircraft noise, road traffic noise, machinery noise reduction, sound intensity measurement techniques, modern instrumentation for noise control and noise regulations are included.

86-1621
Research on a Noise Control Device
K. Mizuno, H. Sekiguchi, K. Iida
Bridgestone Co., Yokohama City, Japan
Bull. JSME, 28 (245), pp 2737-2743 (Nov 1985) 20 figs, 1 table, 4 refs

KEY WORDS: Noise reduction

This noise control device, which produces a greater noise reduction effect than a barrier, is capable of refracting the sound waves and reducing a noise by destruction interference. It consists of many pipes which make a structural delay circuit. It is very important to design the pipes of the noise control device such that the device may produce a great noise reduction effect when it is applied to noise pollution problems. This paper describes the fundamental design method of the pipes and refers to the experimental data in an anechoic room and some sound theories. As a result of this study, the fundamental design method of this noise control device has been established.

86-1622
Determination of Sound Reduction Indices in the Presence of Flanking Transmission
B.G. van Zyl, P.J. Erasmus, G.J.J. van der Merwe

National Physical Res. Lab., Pretoria, Rep. of South Africa
Appl. Acoust., 19 (1), pp 25-39 (1986) 6 figs, 3 tables, 2 refs

KEY WORDS: Noise reduction, Acoustic intensity method

Unlike the standard sound insulation test method, which requires diffuse sound field in both source room and receiving room, the sound intensity method only requires the source room to be diffuse. Flanking may cause large errors in the sound reduction indices obtained by the standard method. The flanking power should be measured separately and subtracted from the total power. The sound intensity method allows selective determination of sound reduction indices for test objects surrounded by flanking surfaces. The measurement surface in the receiving room must be defined in such a way that it encloses nothing but the test object.

86-1623
Reduction of Artillery Noise by Natural Barriers
R. Raspet
US Army Const. Engrg. Res. Lab., Champaign, IL
Appl. Acoust., 19 (2), pp 117-130 (1986) 3 figs, 5 tables, 10 refs

KEY WORDS: Noise reduction, Artillery fire, Noise barriers

Measurements of the reduction is peak sound pressure level of artillery noise due to diffraction over hills have been obtained and compared to the results of a pulse diffraction theory. It is found that Kirchoff boundary conditions most accurately predict the results. The use of a two-barrier model is investigated. The measurements necessary to resolve the questions raised by this study are also described.

86-1624
Acoustical Properties of Plastic Foams with High Specific Flow Resistance (Proprietes acoustiques des mousses a forte resistance specifique au passage de l'air)
J.F. Allard, A. Aknine, A.M. Bruneau
Laboratoire d'Acoustique, Le Mans, Cedex, France
Acustica, 59 (2), pp 142-147 (Dec 1985) 3 figs, 5 tables, 13 refs (in French)

KEY WORDS: Foams, Acoustic properties, Biot theory

The acoustical properties of plastic foams with high specific flow resistance have been modeled by using a simplified version of the Biot's theory. A new method for measuring impedance has been worked out in order to measure the normal impedance of these foams in a free-field. The theoretical predictions of the Biot theory and the measurements performed on a sample are in good agreement. It has been pointed out that the real and the imaginary part of the dynamic shear modulus of the structure can be evaluated by acoustical measurements of these foams.

86-1625
Note on the Relationship of Sound Pressure and Sound Intensity
H. Bloemhof
Brown Boveri & Cie, Baden, Switzerland
Appl. Acoust., 19 (3), pp 159-166 (1986) 3 figs, 3 refs

KEY WORDS: Sound pressure levels, Acoustic intensity method

It is usual to calculate the sound pressure at a given location by energy addition of all relevant components. Another approach considers the sound power per unit area. This note shows that the two approaches lead to different results for ideal line or planar sources. The calculated sound pressure level is higher than expected on the basis of sound intensity. For this reason, the use of sound pressure levels measured close to a source leads to a sound power estimate which is too high.

86-1626
Reverberation Effects on Directionality and Response of Stationary Monopole and Dipole Sources in a Wind Tunnel
K.J. Baumeister
NASA, Lewis Res. Ctr., Cleveland, OH
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 82-90 (Jan 1986) 14 figs, 14 refs

KEY WORDS: Acoustic tests, Wind tunnel testing

Analytical solutions for the three-dimensional inhomogeneous wave equation with flow in a hardwall rectangular wind tunnel and in the free field are presented for a stationary monopole noise source. Dipole noise sources are calculated by combining two monopoles 180 deg out of phase. Numerical calculations for the modal content, spectral response and directivity for both monopole and dipole sources are presented. In addition, the effect of tunnel alterations, such as the addition of a mounting plate, on the tunnels reverberant response are considered. In the frequency range of practical importance for the turboprop response, important features of the free field directivity can be approximated in a hardwall wind tunnel with flow. The major lobe of the noise source must not be directed upstream. However, for an omnidirectional source, such as a monopole, the hardwall wind tunnel and free field response will not be comparable.

86-1627

Spectral Analysis of Numerical Solutions to the Burgers-Korteweg-DeVries Equation

K.V. Rao
David W. Taylor Naval Ship Res. and Dev. Ctr., Bethesda, MD

J. Acoust. Soc. Amer., 79 (1), pp 26-30 (Jan 1986) 8 figs, 11 refs

KEY WORDS: Sound waves, Wave propagation

Finite amplitude acoustic wave propagation through bubbly liquids is studied by using the Burgers-Korteweg-DeVries model equation. Numerical solutions are obtained using a pseudo-spectral method. Spectral analysis of the numerical solutions is performed to study the spectral energy transfer due to nonlinearity. Numerical solutions of the two limiting cases obtained by neglecting either the dissipation or dispersion term of the Burgers-Korteweg-Devries equation are also analyzed. In nonlinear wave propagation, short waves are generated by long waves due to nonlinearity. In bubbly media, these short waves may undergo strong resonance absorption due to the presence of bubbles, even if other mechanisms of dissipation are negligible. The effect of such an absorption is simulated by applying a low-pass filter on the results obtained with the dissipation term neglected. The filter eliminates the short waves generated after each time step. It is shown that such resonance absorption corrections may be necessary for any quantitative comparisons of computed results with experiment.

86-1628

A Simplified Method of Predicting the Output Probability Distribution Based on the Time Series Model for Actual Sound Insulation Systems with Arbitrary Random Excitation by Use of the Statistical Method of Moment

M. Ohta, H. Yamada
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Appl. Acoust., 19 (1), pp 1-15 (1986) 3 figs, 1 table, 4 refs

KEY WORDS: Acoustic insulation, Parameter identification techniques, Statistical analysis

This paper presents a fairly simplified identification method applicable to the actual situation of a sound insulation system under the introduction of a time series system model as seen from the probabilistic viewpoint. This identification method is based on the statistical data obtained from a subexperiment with use of a typical white noise input and the well-known moment method. Then, on the basis of these identified results, a simple method is proposed for predicting the output probability distribution of the insulation system with an arbitrary noise excitation. Finally, the proposed identification and prediction method is applied to several types of actual sound insulating systems and its effectiveness confirmed experimentally.

86-1629

Broadband Noise Propagation in a Pekeris Waveguide

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Instituto de Pesquisas da Marinha, Arraial do Cabo, Brazil

J. Acoust. Soc. Amer., 79 (1), pp 76-83 (Jan 1986) 8 figs, 20 refs

KEY WORDS: Underwater sound, Sound waves, Wave propagation

A theory of broadband noise propagation from a point source in shallow water is developed by use of a Pekeris model with a lossy bottom. The broadband interference pattern in the poser spectrum is calculated analytically and numerically including both the proper mode and the virtual mode fields. It is shown that the interference maxima of the proper mode spectrum describe straight lines in the range-frequency plane and that the virtual mode fields (branch-line integral) are localized. They decay slowly with a range around the cutoff frequencies of the proper modes. It is also shown that bottom attenuation does not influence the form of the power spectrum in any significant way up to ranges of 100 water depths.

86-1630

Low-frequency Rolloff in the Response of Shallow-Water Channels

P.W. Smith, Jr.

Bolt Beranek and Newman, Inc., Cambridge, MA
J. Acoust. Soc. Amer., 79 (1), pp 71-75 (Jan 1986) 5 figs, 1 table, 8 refs

KEY WORDS: Underwater sound, Sound waves, Wave propagation

Propagation in a shallow-water channel of constant sound speed, overlying a homogeneous viscoelastic half-space with frequency-independent loss factors, is examined as a function of range, frequency, and water depth. In addition, a parameter representing the initial rate of increase of bottom reflection loss with increasing grazing angle is examined. Particular attention is focused on the transition frequency below which the transmission loss increases rapidly. It is found to separate a ray-theoretical domain, where the directional spectrum of transmitted sound is quasicontinuous, from a modal domain where, in fact, only the first mode carries significant energy.

86-1631
Intensity Decay Laws for Near-Surface Sound Sources in the Ocean
R.N. Denham
Ministry of Defence, Auckland, New Zealand
J. Acoust. Soc. Amer., 79 (1), pp 60-63 (Jan 1986) 2 figs, 23 refs

KEY WORDS: Underwater sound, Acoustic intensity method

Expressions derived for the acoustic intensity decay with range in shallow water by Marsh and Schulkin have been adapted to describe the intensity-range variation in the deep water situation. The sound speed at the surface is equal to the sound speed at the base of the water column. The modified equations take account of the surface decoupling loss which occurs at frequencies below 100 Hz for source depths less than 25 m. The levels predicted by these equations are compared with experimental data obtained in deep water in the Indian Ocean. It is found that the mean difference in measured and predicted levels at frequencies of 25 and 50 Hz is less than 2 dB, and rms level differences are within 4 dB.

86-1632
Measurement of Down-Slope Sound Propagation from a Shallow Source to a Deep Ocean Receiver
W.M. Carey
NORDA, Arlington, VA
J. Acoust. Soc. Amer., 79 (1), pp 49-59 (Jan 1986) 11 figs, 30 refs

KEY WORDS: Sound waves, Wave propagation, Ship noise, Underwater sound, Experimental data

An experiment has been performed to investigate the coupling of surface-ship noise to the deep ocean sound channel. These calibrated measurements of sound propagation from the 19-m, 135-Hz source were obtained at a deep ocean receiver while the source tow proceeded from deep water up the Sable Island Bank at ranges between 730 and 910 km. The sound propagation path was from the cold slope waters over the bank through the Gulf Stream and to the edge of the Sargasso Sea. The mean value of the transmission loss was 100 dB with a down-slop enhancement estimated to be 4 dB resulting from the combined effects of trapping in the strong shallow sound channel and reflections from the slope. Comparisons with calculated results using the parabolic equation method were good and demonstrate coupling to the deep ocean sound channel.

SHOCK EXCITATION

86-1633
Decoupling of Secondary Systems for Seismic Analysis
A.H. Hadjian, B. Ellison
Bechtel Power Corp., Norwalk, CA
J. Pressure Vessel Tech., Trans. ASME, 108 (1), pp 78-85 (Feb 1986) 8 figs, 3 tables, 8 refs

KEY WORDS: Seismic analysis, Equipment response

Interest in the decoupling of such secondary systems as floor-mounted equipment centers on two decoupled models, the cascading model and the lumped model. A set of decoupling criteria is derived which addresses both models. The solution is based on the two-degree-of-freedom system. Error in predicted frequency given by each decoupled model is presented. Similar results for response error are derived based on broad-band input to the supporting system. The broad-band input would be applicable to subsystems supported by primary structures. Recommendations are made for extension of the problem to arbitrary multiple-degree-of freedom systems. The findings of this study will provide the analyst with the information necessary to choose the best decoupled model and will indicate whether decoupling should be attempted.

86-1634
Inelastic Earthquake Response of Asymmetric Structures
Y. Bozorgnia, W.K. Tso
ASCE J. Struc. Engrg., 112 (2), pp 383-400 (Feb 1986) 14 figs, 12 refs

KEY WORDS: Seismic response, Asymmetric structure, Torsional response

The inelastic seismic response of a class of one-way torsionally unbalanced structures is presented. The structural model consists of a single mass supported by bilinear hysteretic elements. The yield strength of the model is a function of the lateral period and follows a trend similar to the design strengths suggested by design codes. The response parameters of interest are the resisting element ductility demands and the edge displacement of the model. The sensitivity of the response parameters to system parameters, such as structural eccentricity, yield strength, uncoupled lateral period, and uncoupled torsional to lateral frequency ratio, is examined for two types of ground motions. It is found that the effect of asymmetry is most pronounced for stiff structures with low yield strength. Unlike elastic response, the uncoupled torsional-to-lateral-frequency ratio is not a sensitive system parameter that affects the inelastic response. Exceptionally large ductility demand can be expected on eccentric stiff structures with low yield strength when exposed to ground motions with large and long duration acceleration peaks as exemplified by the 1977 Romanian earthquake record.

86-1635
Stochastic Variation of Earthquake Ground Motion in Space and Time
R.S. Harichandran, E.H. Vanmarcke
Michigan State Univ., East Lansing, MI
ASCE J. Engrg. Mech., 112 (2), pp 154-174 (Feb 1986) 13 figs, 1 table, 11 refs

KEY WORDS: Seismic excitation, Ground motion, Stochastic processes

Consideration of the spatial as well as temporal variation of earthquake ground motions may be important in the design of structures with spatially extended foundations and lifeline systems. In this paper, the ground motion during a specific earthquake event is conceived as the realization of a space-time random field. Following a brief review of pertinent results from the second-order theory of random fields and a description of spectral estimators, accelerograms recorded by the SMART 1 seismograph array in Lotung, Taiwan, are examined to determine the frequency-dependent spatial correlation of earthquake ground motions. A scheme for processing seismograph array data is proposed, and similarities and differences in the motions recorded during different events are noted. Finally, key parameters characterizing the correlation structures are identified and a preliminary mathematical model is suggested for the space-time correlations.

86-1636
Limiting Performance of Shock Isolation Systems by a Modal Approach
W.D. Pilkey, L. Kitis
Univ. of Virginia, Charlottesville, VA
Earthquake Engrg. Struc. Dynam., 14 (1), pp 75-81 (Jan-Feb 1986) 1 fig, 2 tables, 7 refs

KEY WORDS: Optimization, Shock isolators, Modal analysis, Linear programming

The problem of determining the limiting performance of vibrating systems under shock loading is solved by replacing portions of the system by control forces which can represent any design. For the class of problems treated here, the performance index and the constraints are linear combinations of system response variables such as displacements, velocities and accelerations. Furthermore, the equations of motion are linear, so that is is possible to formulate the optimization procedure as a linear programming problem. In expressing the performance index and the constraints as linear functions of the unknown control forces, a modal approach is used to simplify and improve previous treatments of this problem. In spite of these linearity requirements, the control forces are not required to be linear functions of the response variables.

86-1637
Study of Collapse of a Free Surface Conical Cavity Due to a Plane or Spherical Shock Wave
H.S. Yadav, N.K. Gupta
IIT Delhi, New Delhi, India
Intl. J. Impact Engrg., 3 (4), pp 217-232 (1985) 12 figs, 1 table, 22 refs

KEY WORDS: Shock waves, Penetration

A plane faced or spherical target has been subjected to plane or spherical shock loading at its other face by contact explosive or the impact of a flyer plate. It has a central conical cavity at one face. The shock wave generated in the target interacts with the cavity and as a consequence the cavity collapses and a high velocity metal jet is produced. Target free surface velocity, shock wave velocity and jet velocity were measured using high speed oscilloscopes. The results revealed that the jet velocity is linearly related to the shock-induced particle velocity in the target. Considering it to be sink-type incompressible flow, a semiempirical analysis of the collapse has been presented which agrees well with the experiments.

86-1638

Large Deformations of Rigid-Viscoplastic Cantilevers Under Impulsive Loading

R. Trossbach, J.B. Martin
Univ. of Cape Town, Rondebosch, Rep. of South Africa
Intl. J. Impact Engrg., 3 (4), pp 243-258 (1985) 9 figs, 1 table, 10 refs

KEY WORDS: Cantilevers, Metals, Impulse response, Mode shapes, Pipe whip

The problem of ductile metal cantilever structure subjected to dynamic loads leading to deformations of the order of the dimensions of the structure is considered. The material is treated as rigid-viscoplastic; in this idealization elastic effects are ignored, and the dependence of the yield stress on the rate of strain is taken into account. Applications of the method are described for impulsive loading, and extended to the pipe-whip problem where the loading is in the form of a pulse which acts in the direction of the tangent at the tip of the cantilever structure at each instant.

86-1639

The Transient Stress in an Elastic Half-space Excited by Impulsive Loading Over One Quarter of its Surface

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Gunma Univ., Kiryu City, Japan
Bull. JSME, 29 (247) pp 44-51 (Jan 1986) 9 figs, 4 refs

KEY WORDS: Transient response, Elastic half space, Stress waves, Wave propagation

The propagation of transient waves in an elastic half space excited by a normal or tangential impulsive loading over one quarter of its surface is investigated. The solution is based on the stress function approach and the Laplace transform, the double Fourier transform method is applied to a non-axisymmetric dynamical problem. The stress transient in load region was compared with that on free surface. The numerical results are shown graphically for the stress variations versus time.

86-1640

On Elastic-Plastic and Rigid-Plastic Dynamic Response with Strain Rate Sensitivity

J.M. Mosquera, P.S. Symonds, H. Kolsky
Brown Univ., Providence, RI
Intl. J. Mech. Sci., 27 (11/12), pp 741-749 (1985) 12 figs, 20 refs

KEY WORDS: Pulse excitation, Elastic plastic properties, Rigid plastic properties

A method is described for calculating the response of one-dimensional structures to arbitrary pulse loading, which takes account of strain rate sensitivity in an approximate but realistic manner. The method assumes the flow stress to depend on plastic strain rate according to an overstress power law. The difficulty of relating dynamic flow stress to velocity is overcome by introducing a factor related to the length of the plastic hinge zone, which is evaluated by numerical experiments for a basic response pattern.

86-1641

The Impact End Stress of a Bar Subjected to Longitudinal Compression Impact

M. Naitoh, M. Daimaruya, K. Liu
Muroran Inst. of Technology, Hokkaido, Japan
Bull. JSME, 28 (245), pp 2585-2591 (Nov 1985) 9 figs, 20 refs

KEY WORDS: Cylinders, Bars, Impact response

This paper experimentally and theoretically investigates the elastic-plastic stress at the impact end of a cylindrical specimen subjected to longitudinal compression impact with a stress bar which remains elastic during the test. The measured elastic response of the stress bar showed explicitly an elevation of dynamic stress and its relaxation at the impact end of the specimen. The impact end stress of the specimen was analyzed by using the strain-rate dependent theory for plastic wave propagation, taking into account a rise time of impact and impact conditions. An extreme elevation of the dynamic stress was predicated in the case of a step impact, but it went down rapidly with an increase in the rise time of an impact velocity. Taking the stress as an incident stress to the stress bar, the elastic response of the stress bar based on the Love theory for elastic waves almost agreed quantitatively with the experimental results as well as qualitatively.

86-1642
Analysis of Nonlinear Plane Wave Propagation in Solids (Méthodes d'analyse de la propagation non-linéaire d'ondes planes dans un solide)
M. Planat, E. Francois
Laboratoire de Physique et Métrologie des Oscillateurs du C.N.R.S., Besançon, France
Acustica, 59 (2), pp 102-111 (Dec 1985) 1 fig, 25 refs (in French)

KEY WORDS: Wave propagation, Shock waves, Elastic media

Nonlinear plane wave propagation in an homogeneous elastic solid is studied. First the characteristic method is used to explain the simple wave and shock-wave formation from an arbitrary excitation condition. An implicit solution is obtained in the case of the simple wave, which is developed for both cases of the near- and far-field from the excitation source. Then various approximation techniques are tested in comparison with the previous exact approach: the direct successive approximation method, a multiple scale method and a coupled amplitude theory. This comparison is of interest to solve more complex nonlinear problems by means of such resolution.

86-1643
Bounds on the Dynamic Plastic Behaviour of Structures Including Transverse Shear Effects
N. Jones
Univ. of Liverpool, Liverpool, UK
Intl. J. Impact Engrg., 3 (4), pp 273-291 (1985) 10 figs, 24 refs

KEY WORDS: Beams, Plates, Shells, Dynamic plasticity

The influence of transverse shear is an important factor in the dynamic plastic response of several practical problems. In fact, transverse shear effects play a vital role in the severance of structures at hard points and dominate the response of ideal fiber-reinforced structures as well as being potentially important for higher modal structural responses. It is demonstrated in this article that the simple bound theorems, which were developed for rigid plastic continua, can provide excellent estimates of the response durations and permanent displacement of impulsively loaded beams, circular plates and cylindrical shells when transverse shear effects are important. The bound theorems also provide exact agreement with the dynamic behavior of the ideal fiber-reinforced structures which have been examined.

VIBRATION EXCITATION

86-1644
Composite Statistical Method for Modeling Wind Gusts
J.R. Schiess
NASA Langley Research Center, Hampton, VA
J. Aircraft, 23 (2), pp 131-135 (Feb 1986) 7 figs, 5 refs

KEY WORDS: Wind induced excitation, Aircraft

This paper discusses the application of three statistical methods in combination to model wind gusts for use in aircraft flight simulation. The approach combines principal components analysis, time-series analysis, and probability distribution methods to analyze and simulate wind gust components. Comparisons between wind gust components generated by the model and those measured onboard an aircraft show the model produces realistic gust components.

86-1645
Simulation of Random Surface Roughness-Induced Contact Vibrations at Hertzian Contacts During Steady Sliding
A. Soom, J.-W. Chen
State Univ. of New York, Buffalo, NY
J. Trib., Trans. ASME, 108 (1), pp 123-127 (Jan 1986) 10 figs, 11 refs

KEY WORDS: Contact vibrations, Surface roughness, Hertzian contact

Random normal contact vibrations, excited by surface irregularities swept through the contact region of Hertzian contacts during sliding, are studied using digital simulation techniques. The input disturbances are modeled as random time processes with specified spectral content in the spatial wavenumber and frequency domains. The Hertzian contact stiffness is modeled directly or through a bilinear approximation. The contact

vibration spectra and resulting mean square contact loading are obtained from the simulations. A comparison with previous measurements shows good agreement between the simulation and experimental results.

86-1646
Dynamic Response of Linear Structures to Random Trains of Impulses with Random Spatial Shapes
R. Iwankiewicz
E.N.S.E.T./Universite Paris-VI C.M.R.S., Cachan, France
J. de Mécanique Théor. Appl., 4 (4), pp 485-504 (1985) 5 figs, 16 refs

KEY WORDS: Random excitation, Normal modes, Modal analysis

An analytical technique is developed to determine the dynamic response of linear structures to random impulses with random spatial shapes. The train of impulses is characterized in time domain by the stochastic point process and in space domain the impulses are randomly located and have random shapes. The concept of the doubly stochastic point process is used which allows to consider the general situation, when the arrival times and locations of impulses are correlated. The normal mode approach is used and the general expressions are provided for mean values and cross-correlations of the modal responses. The contribution of modal responses in the mean-square response of shallow cylindrical shells to random impulses with white-noise stochastic spatial shape is discussed and compared with the case of uniform deterministic spatial distribution of the excitation.

86-1647
Unsteady Flow Near a Circular Cylinder Oscillated Sinusoidally in Uniform Flow (1st Report, Vortex-shedding Mechanism in Synchronization Phenomena)
H. Nagata, K. Hayashi, M. Kawai
Gifu Univ., Gifu, Japan
Bull JSME, 28 (246), pp 2931-2939 (Dec 1985) 11 figs, 11 refs

KEY WORDS: Circular cylinders, Vortex shedding

The flow past a circular cylinder undergoing transverse oscillation in a uniform flow is investigated by means of flow visualization technique. Oscillations of the separation points and the stagnation point, and the rate of vorticity fluxes through the section near the separation points where examined during a cycle. The phenomenon that the vortex sheds periodically in the same frequency as the cylinder was found to be caused by a significant change in the vorticity distribution in the vortex region. It was induced by vorticity feeding from the third separation point found behind two ordinary separation points.

86-1648
Chaotic Behavior in a System with Hysteresis
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Shanghai Jiaotong Univ.
Acta Mech. Solida Sinica, 3, pp 296-305 (1985) 6 figs, 7 refs

KEY WORDS: Oscillators, Hysteretic damping

The steady motion of a nonlinear oscillation system exhibits periodic, almost periodic and chaotic behavior which can be observed by a spectral and other analyses of the Poincare set. A nonlinear oscillation system is discussed, in which the relation of the restoring force to the displacement is in the form of a hysteresis loop. The self-sustained oscillation of the system is investigated with the analytical method. Apart from a lot of periodic solutions with various periods, some solutions with a quasi-periodic/chaotic behavior are also found.

86-1649
Dynamics of Two Strongly Coupled Relaxation Oscillators
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Univ. of Washington, Seattle, WA
SIAM J. Appl. Math., 46 (1), pp 56-67 (Feb 1986) 6 figs, 20 refs

KEY WORDS: Oscillators, Van der Pol method

This paper concerns the dynamics of a pair of identical, linearly coupled van der Pol relaxation oscillators. The stability of the in-phase and out-of-phase modes of vibration is studied. The stability of both modes is shown to be governed by the behavior of a linear variational equation with periodic coefficients. Approximate analytical solutions are obtained by the method of matched asymptotic expansions. These analytical results are supplemented by numerical integrations based on Floquet theory. It is shown that previous work based on the sinusoidal (nonrelaxation) limit fails to predict a significant region of instability for both modes.

86-1650
Free Vibration of a Rectangular Parallelepiped Using the Theory of a Cosserat Point
M.B. Rubin
Technion -- Israel Institute of Technology, Haifa, Israel
J. Appl. Mech., Trans. ASME, 53 (1), pp 45-50 (Mar 1986) 2 figs, 8 refs

KEY WORDS: Parallelepiped bodies, Cosserat point, Longitudinal vibration, Shear vibration, Natural frequencies

Free vibration of a rectangular parallelepiped composed of a homogeneous linear elastic isotropic material is studied. The parallelepiped is modeled as an isotropic Cosserat point and simple formulas are developed to predict the lowest frequencies of vibration. Within the context of the theory, extensional and shear vibrations are uncoupled. The lowest extensional frequency predicted by the Cosserat theory is compared with available exact solutions and with predictions of thin rod theory. Finally, by introducing a simple modification of the director inertia coefficient it is shown that the Cosserat predictions of the extensional frequencies are correct.

86-1651
Dynamic Analysis of Constrained System of Rigid and Flexible Bodies with Intermittent Motion
Y.A. Khulief, A.A. Shabana
Univ. of Illinois Chicago Circle, Chicago, IL
J. Mech., Transm., Autom. in Des., 108 (1), pp 38-45 (Mar 1986) 13 figs, 16 refs

KEY WORDS: Intermittent motion, Constrained structures

A method for dynamic analysis of large-scale constrained system of mixed rigid and flexible bodies with intermittent motion is presented. The system equations of motion are written in the Lagrangian formulation using a finite set of coupled reference position and local elastic generalized coordinates. Equations of motion are computer generated and integrated forward in time using an explicit-implicit integration algorithm. Points in time at which sudden events of the intermittent behavior occur are monitored by an event predictor which controls the integration algorithm and forces a solution for the system impulse-momentum relation at those points. Solutions of impulse-momentum relations define the jump discontinuities in the composite velocity vector as well as the generalized impulses of the reaction forces at different joints of the mechanical system.

MECHANICAL PROPERTIES

DAMPING

86-1652
Characteristics of Squeeze Film Dampers (Kennlinien realer Quetschöldämpfer)
J. Glienicke, M. Schwer
Konstruction, 37 (8), pp 301-308 (Aug 1985) 11 figs, 11 refs (in German)

KEY WORDS: Squeeze film dampers

The dynamic characteristics of squeeze film dampers, used to improve rotor dynamics, are determined theoretically and experimentally. Damping capacity of squeeze-film dampers depends on the basic parameters: diameter, width, viscosity and clearance as well as design of the oil inlet dimensions and the inertia forces of oil. The dynamic load limit depends on the geometry of oil feed and the pressure of the supplied oil.

86-1653
Study on an Oil Damper with Variable Damping Mechanism
T. Asami, H. Sekiguchi, S. Taniguchi
Himeji Inst. of Technology, Hyogo, Japan
Bull. JSME, 28 (246), pp 2978-2985 (Dec 1985) 10 figs, 2 tables, 6 refs

KEY WORDS: Oil dampers

This work is on the design of a variable oil damper whose damping is adjustable by varying the area of orifice in the piston. Two new methods for analyzing the damper are proposed. The first method of analysis is based on the assumption that the diameter of the piston is equal to the inner diameter of the cylinder but no friction occurs between the contact surfaces.

86-1654
Bounding Theorems of Complex Eigenfrequencies
Chen Shao-ting
Acta Mechanica Sinica, 17 (4), pp 328-339 (1985) CSTA No. 531-85.56

KEY WORDS: Modal damping, Damping coefficients

Two bounding theorems with regard to regions containing complex eigenfrequencies for vibrating systems with damping have been proved. From these theorems, a formula for the bounding of damping coefficients corresponding to complex mode of vibration and a new formula calculating the modal damping coefficients are presented.

86-1655
An Analytical Model for the Vibration of Laminated Beams Including the Effects of Both Shear and Thickness Deformation in the Adhesive Layer
R.N. Miles, P.G. Reinhall
Univ. of Washington, Seattle, WA
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 56-64 (Jan 1986) 6 figs, 14 refs

KEY WORDS: Viscoelastic damping, Viscoelastic-core-containing media, Beams, Layered materials, Stiffness effects

The equations of motion governing the vibration of a beam consisting of two metal layers bonded together with a soft viscoelastic damping adhesive are derived and solved. The adhesive is assumed to undergo both shear and thickness deformations during the vibration of the beam. In previous investigations the thickness deformation has been assumed to have negligible effect on the total damping. However, if the adhesive is very soft, and if at least one of the metal layers is stiff in bending, the thickness deformation in the adhesive can become the dominant damping mechanism. The analysis presented here comprises an extension of the well-known sixth order theory of DiTaranto, Mead, and Markus to include thickness deformation.

86-1656
Control Effects of Damped Sandwich Beam on Random Vibration
Xi Dechang, Chen Qinghua, Cai Guoqiang
Zhejiang Univ., People's Rep. China
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 65-68 (Jan 1986) 8 figs, 12 refs

KEY WORDS: Damping effects, Viscoelastic damping, Beams, Sandwich structures, Random response

This paper investigates the control effects of sandwich beam to random excitation. The statistical properties of the transverse displacement and stress are analyzed. The control effects of the damping layer (including the viscoelastic layer and the constraining layer) on random transverse displacement and random stress are investigated. The results for a sandwich beam are compared with those for the original beam. Finally, examples are given to illustrate the method.

FATIGUE

86-1657
Fatigue Crack Growth from the Standpoint of Crack Energy Density
K. Watanabe
Univ. of Tokyo, Japan
Bull. JSME, 28 (245), pp 2511-2518 (Nov 1985) 9 figs, 9 refs

KEY WORDS: Fatigue life, Crack propagation

The crack energy density was proposed by the author as one of the most important crack parameters, of which the physical meaning is clear throughout the life of a crack. In this paper, this crack energy density is applied to the problem of fatigue crack growth. The role of crack energy density as a crack parameter is made clear. The relationship between the range of crack energy density and the load-displacement curves of specimens is obtained. It is shown that the growth rate of fatigue crack is determined by the increment of crack energy density per one cycle.

86-1658
Fatigue Analysis of Plain Concrete in Flexure
B.H. Oh
ASCE J. Struc. Engrg., 112 (2), pp 273-288 (Feb 1986) 6 figs, 3 tables, 32 refs

KEY WORDS: Fatigue life, Concrete, Bridges, Pavements

This paper involves fatigue strength of plain concrete subjected to flexural loading. This type of fatigue loading is of concern in the design of concrete bridges and concrete pavement slabs. The flexural stresses in these structures can be critical. Both experimental and theoretical studies are conducted. The S-N curves are generated from the test results and an equation is obtained by regression analysis to predict the flexural fatigue strength of concrete. A probabilistic approach is introduced to predict the fatigue reliability of concrete. The method of

obtaining the distribution parameters from S-N relation for the flexural fatigue of concrete is presented. The Weibull distribution is found to have more convincing physical features than the lognormal distribution, and may be appropriate to describe the fatigue behavior of concrete.

86-1659
Fatigue of Partially Prestressed Concrete
M. El Shahawi, B. deV. Batchelor
Queens Univ., Kingston, Ontario, Canada
ASCE J. Struc. Engrg., 112 (3), pp 524-527 (Mar 1986) 10 figs, 2 tables, 24 refs

KEY WORDS: Fatigue life, Beams, Prestressed concrete, Cracked media

The fatigue behavior of partially prestressed concrete beams is investigated in a series of tests of bonded post-tensioned T-beams. The beams are simply supported with the same overall dimensions, and designed for the same flexural strength. The main parameter in the study is the degree of prestress. Eight beams are tested in fatigue: five beams are subjected to a constant load cycle, while the remaining three beams are subjected to cumulative fatigue loading. All beams are initially cracked before the application of repeated loading. Rapid changes in cracking, deflections, and non-prestressed steel stress occur in the early stages of fatigue loading, and are followed by a stable period until just prior to failure. Cumulative fatigue loading always results in reduced life of the member. Fatigue failure is caused by successive fracturing of the non-prestressed reinforcement; and no failure of pretensioned reinforcement is observed. Crack spacing approximately coincides with shear reinforcement spacing which is constant in the tests, indicating that the latter spacing may have some effect on spacing of cracks. Based on the test results, it is concluded that at present there is no justification for the use of complicated expressions to predict maximum crack width. The simple expression proposed by the CEB-FIP Code gives satisfactory predictions.

ELASTICITY AND PLASTICITY

86-1660
Elastoplastic Analysis Method for Dynamic Agencies
C. Polizzotto
Univ. of Palermo, Palermo, Italy
ASCE J. Engrg. Mech., 112 (3), pp 293-310 (Mar 1986) 7 figs, 39 refs

KEY WORDS: Elastic-plastic properties, Modal superposition method

A method for the analysis of discrete elastoplastic structural systems with proportional damping and subjected to dynamic laos is presented. Firstly, it is shown that, using the mode superposition method, the actual node displacements and generalized stresses can be given suitable integral representations. Secondly, considering the evolution of the system in a small time step and imposing the relevant laws of (holonomic) plasticity theory at the step extreme times only, it is shown that the step plastic strain increments can be obtained by solving algebraic problems shaped either in the form of a linear complementarity problem, or in the forms of two alternative quadratic programming problems. The formats of these problems are recursive and thus sequentially applicable, such as to cover the entire loading process, in much the same way quasistatic elastoplastic problems can be solved. The resulting numerical procedure turns out to be unconditionally stable, while the only error sources are due to the modeling of the unknown plastic strain history. Imposed strain-like loads (e.g., thermal shocks) can be considered. A simple numerical example is also presented and a preliminary comparison of the proposed method with other known methods is discussed.

WAVE PROPAGATION

86-1661
Coupled Mode Theory of Intrinsic Modes in a Wedge
J.M. Arnold, L.B. Felsen
Univ. of Glasgow, Glasgow, Scotland
J. Acoust. Soc. Amer., 79 (1), pp 31-40 (Jan 1986) 3 figs, 11 refs

KEY WORDS: Sound waves, Wave propagation

Recent theoretical studies of acoustic wave propagation in a model waveguide consisting of a homogeneous wedge with one reflecting and one penetrable boundary have established the utility of the concept of an intrinsic mode. It generalizes for nonseparable problems the normal mode of separable configurations. This paper shows by explicit calculations on the same model problem that direct orthogonal expansion (in local normal modes) of the intrinsic mode preduces the same expansion coefficients as those obtained by a perturbation analysis of the coupled mode equations. The perturbation method used is that of renormalization, in which the mode coupling operator is iteratively diagonalized up to a certain order in the nonseparability parameter, which, in this case, is the wedge angle.

86-1662

Harmonic Wave Propagation in Nonhomogeneous Layered Composites

Z.P. Duan, J.W. Eischen, G. Herrmann
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J. Appl. Mech., Trans. ASME, 53 (1), pp 108-115 (Mar 1986) 8 figs, 10 refs

KEY WORDS: Composite materials, Harmonic waves, Wave propagation

A new method for analyzing plane wave propagation in a periodically layered, elastic, nonhomogeneous composite body is proposed. The nonhomogeneity considered is a variation of the material properties within each composite layer. Results from probability theory are used to arrive at the two fundamental solutions of the governing second order ordinary differential equations. Floquet's wave theory is combined with a Wronskian formula to yield the dispersion relationship for this nonhomogeneous composite. Numerical results show that the presence of material nonhomogeneity affects the range of frequencies which can pass through the composite unattenuated.

EXPERIMENTATION

MEASUREMENT AND ANALYSIS

86-1663

Approximate Structural Response Characterization Using Parametric and Envelope Models for Multimodal Systems

P. Davies, J.K. Hammond
Univ. of Southampton, Southampton, England
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 39-43 (Jan 1986) 8 figs, 8 refs

KEY WORDS: Approximations methods, Frequency response function, Time domain method

In the study of the response of systems to an excitation, there are circumstances when it is desirable to obtain some overall or average characterization of the system and its response rather than a detailed description. In this paper two methods are used to describe the overall features of the system. One is appropriate for the frequency domain and one for the time domain. For modally dense systems the main features of the frequency response function are described in terms of low-order parametric models. While these models may be adequate for the frequency domain representation, they may not produce a good approximation to the response of the system in the time domain. The second approach relates the envelope of the input signal to the envelope of the response signal, in order to describe the overall time domain response characteristics.

86-1664

The Accuracy of Frequency Response Function Measurements Using FFT-Based Analyzers with Transient Excitation

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J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 44-49 (Jan 1986) 9 figs, 6 refs

KEY WORDS: Frequency response function, Fast Fourier transform, Impact excitation

The accuracy of the frequency response measurement obtained using impact excitation and a Fast Fourier Transform based spectrum analyzer has been investigated. It has been shown that with impact excitation, provided the impacts are reproducible and the extraneous noise level is low, the coherence estimates obtained from the analyzer are unity, irrespective of the frequency resolution so unity coherence does not necessarily imply accurate results. The results with impact excitation are compared with those obtained using random excitation where both the coherence and frequency response function estimates are affected by bias error. The theoretical predictions have been verified by tests on an analogue computer and on a built-up structure.

86-1665

Determination of Modal Deformations by Means of In Situ Accelerations

S. Braun, J. Levran

Technion -- Israel Institute of Technology, Haifa, Israel
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 32-38 (Jan 1986) 11 figs, 2 tables, 6 refs

KEY WORDS: Measurement techniques, Mode shapes, Machine tools, Drills

Described is a method of extracting modal deflection patterns by means of acceleration measurements only. This enables the testing of systems/machines under normal operating conditions with many potential advantages. A single inertance measurement can be used in conjunction with these results to estimate true modal parameters. The method is demonstrated for a drill under machining conditions. Error associated with the proposed method are discussed, and estimated for the given example.

86-1666
A New Method of Deconvolution in the Time Domain
Fannien Kong
Acta Electronica Sinica, 13 (4), pp 8-13 (1985) CSTA No. 621.381-85.99

KEY WORDS: Fourier transformation, Z-transform, Time domain method

A new method of deconvolution in the time domain is proposed, and compared with two traditional methods of deconvolution (Fourier transform method and Z transform method). The results show that these traditional methods are the special cases of the new method. The new method can overcome the difficulties of using traditional methods to do deconvolution. Only a discrete case is concerned in the paper.

86-1667
Dynamics in Computer Aided Design of Instrumentation (Rechnergestutzte Konstruktion von Gerategestellen unter Berucksichtigung der Dynamik)
W. Pech
Feingeratetechnik, 35 (1), pp 18-19 (1986) 2 figs, 1 table, 7 refs (in German)

KEY WORDS: Instrumentation, Computer programs

In the dynamic analysis of instrumentation only fundamental modes of vibration need to be investigated. Higher modes are not excited by typical excitation. In the construction of precision instrumentation the characteristic criteria is stiffness. Thus a lumped parameter model is used in which rigid bodies are connected by elastic and damping elements. Using this model differential equations of motion are derived and solved using the BEKOS (calculation of coupled vibrations) computer program.

86-1668
Validity of Intensity Measurements in Partially Diffuse Sound Field
S. Gade
Bruel & Kjaer Instruments, Inc., Marlborough, MA
Tech. Rev. (B & K), 4 (1984) 9 figs, 26 refs

KEY WORDS: Acoustic intensity method

In this article a practical method is proposed and outlined for determining the dynamic capability of intensity analyzing systems and the Reactivity Index of intensity measurements. Furthermore, using this method, the amount of error due to phase mismatch, the amount of random error, and the useful frequency range for measuring intensity in different types of sound field can be determined.

86-1669
Influence of Tripods and Microphone Clips on the Frequency Response of Microphones
K. Zaveri
Bruel & Kjaer Instruments, Inc., Marlborough, MA
Tech. Rev. (B & K), 4 (1984) 10 figs, 1 ref

KEY WORDS: Sound level meters, Supports, Error analysis

Use of microphone clips and tripods to support microphones causes disturbance of the sound field and thus causes errors in sound level measurements. This article illustrates the amount of errors introduced for different mounting configurations, and shows how these errors can be kept to a minimum.

86-1670
The Relationship between Finite Element Analysis and Modal Analysis
N.F. Rieger
Stress Technology Inc., Rochester, NY
S/V, Sound Vib., 20 (1), pp 16-31 (Jan 1986) 22 figs, 6 refs

KEY WORDS: Finite element technique, Modal analysis

The properties and techniques of modal analysis and of finite element analysis are identified, together with the present advantages and shortcomings of both methods. The inter-relationship between these techniques is described, and the contributions of modal analysis to efficient finite element analysis are reviewed. It is noted that the term modal analysis is used to describe a test procedure for obtaining structural data and an analytical procedure for efficient solution of structural and rotor dynamics problems. Four case histories are included which describe the joint use of modal analysis and finite element analysis to diagnose and confirm results obtained in selected practical problems of structural analysis.

86-1671

A Natural Modes Model and Modal Identities for Damped Linear Structures

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J. Appl. Mech., Trans. ASME, 53 (1), pp 33-38 (Mar 1986) 1 table, 16 refs

KEY WORDS: Modal analysis, Modal models, Viscous damping

A modal model is derived for a passive elastic structure with linear viscous damping, from a first-order state variable arrangement of the physical parameters model. Transfer functions and normalizations used in experimental modal parameter estimation are also given special attention.

86-1672

Determination of Dynamic Forces from Wave Motion Measurement

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J. Appl. Mech., Trans. ASME, 53 (1), pp 61-68 (Mar 1986) 14 figs, 1 table, 11 refs

KEY WORDS: Force measurement

An experimental method has been developed for generating oblique forces with known orientations and time histories. Recorded signals from several forces were analyzed by an iterative deconvolution method to determine their orientations and time histories. The recovered values agree closely with the exact ones for these controlled sources. These experiments are a valuable test of source characterization methods that may be applied to seismic data from earthquake sources or to signals recorded from the acoustic emission of cracks.

DYNAMIC TESTS

86-1673

Pseudo-Dynamic Testing of Structures

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ASCE J. Engrg. Mech., 112 (2), pp 183-197 (Feb 1986) 6 figs, 4 tables 26 refs

KEY WORDS: Pseudodynamic testing method, Seismic tests

An experimental procedure is developed for the testing of full- or large-scale structures under simulated seismic deformations in a quasi-static fashion. The procedure, pseudo-dynamic testing, is based on active modal control theory. The pseudo-dynamic testing procedure is a simultaneous simulation and control process in which inertia and damping properties are simulated and stiffness properties are acquired from the structure. The procedure calculates a set of dynamic displacements based on active control theory, utilizing the simulated inertia and damping properties and acquired stiffness properties under a hypothetical ground motion, and simulates the response of the structure under seismic motion in a quasi-static fashion. A detailed implementation procedure is presented for structures that can be modeled as discrete mass systems. The application of the procedure is simulated on the 1/5 scale model of the US-Japan seven-story reinforced concrete frame wall structure.

DIAGNOSTICS

86-1674

Frequency Analysis of the Signals Generated by the Failure of Constituent Wires of Wire Rope

N.F. Casey, H. White, J.L. Taylor

NDT Intl., 18 (6), pp 339-344 (Dec 1985) 11 figs, 4 tables, 5 refs

KEY WORDS: Diagnostic techniques, Wire, Cables, Failure detection, Frequency analysis

Frequency analysis has been applied to the acoustic emission signals resulting from the failure of steel wire ropes and individual wires taken from the ropes. These preliminary tests were performed on notched and unnotched specimens under rising and cyclic loads; the signals were digitized and analyzed using a fast Fourier transform program. Failure of the wires generated signals with frequencies between 0 and 100 kHz; the average peak to peak amplitude values of the signals from individual wires were greater than those from the failure of notched wires within the rope. The signal duration also increased in the rope specimens.

86-1675
Sensitivity of Flexibility Monitoring of Offshore Jacket Platforms
S. Rubin, R.N. Coppolino
The Aerospace Corp., Segundo, CA
J. Energy Resources Tech., Trans. ASME, 108 (1), pp 72-76 (Mar 1986) 5 figs, 2 tables, 9 refs

KEY WORDS: Diagnostic techniques, Drilling platforms, Off-shore structures, Flexibility coefficients

Flexibility monitoring is a vibration-based method for simplifying the detection of major underwater damage on offshore jacket platforms. Ambient vibrations are detected at each of the underwater framing levels relative to abovewater vibration in the fundamental sway and torsional modes. Derived are flexibility parameters which relate to the shear flexibilities of each framing bay and of the foundation. Great promise has been shown by laboratory and field testing. This paper presents a comprehensive sensitivity assessment for severance of diagonal members over a wide range of structural redundancy for generic platform configurations.

86-1676
Preventive Maintenance for Roller and Journal Bearings of Induction Motor Based on the Diagnostic Signature Analysis
T. Koizumi, M. Kiso, R. Taniguchi
Mitsubishi Electric Corp., Amagasaki, Japan
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 26-31 (Jan 1986) 11 figs, 8 refs

KEY WORDS: Diagnostic techniques, Signature analysis, Roller bearings, Journal bearings

This paper is concerned with the preventive maintenance of roller and journal bearings installed in induction motors. Almost all kinds of failure modes happening on both roller and journal bearings have been reproduced and classified using time and frequency domain data analysis. Diagnostic procedure also has been derived using these analyzed results and statistical method. Finally, an actual diagnostic system for the early stage detection of defected roller bearings has been developed for practical field use.

86-1677
Excessive Scrubber Fan Vibration
J.L. Caggiano
Center Engrg. Inc.
Vibrations, 1 (4), p 19 (Mar 1986) 1 fig

KEY WORDS: Diagnostic techniques, Fans, Vibration control, Case histories

A 6000 hp motor-driven scrubber fan tripped due to high vibration. Following coast down, which required about 15 minutes, the unit could be restarted and vibration levels were acceptable. After a few months, however, the frequency of the trips increased. Results confirmed that then unbalance effect closely resembled a speed-squared function. The vibration problem was eliminated by installing a fan wheel with a 0.012 in. interference fit.

86-1678
Identification of Chipped Teeth
D.B. Szrom
Mechanical Consultants Incorporated, Homewood, IL
Vibrations, 1 (4), p 20 (Mar 1986) 3 figs

KEY WORDS: Diagnostic techniques, Gear drives, Gear teeth, Case histories

This case history is concerned with gear reduction units with chipped teeth. The presence of two defects and one pulse can be explained by the geometry of the gear mesh (high contact ratio). Because of the different diameters of the pinion and bull gear, the first defect exits the load zone as the second defect enters. Both are in mesh at the same time. The result is a single pulse with high harmonic content in the frequency domain. The pulse in the time domain and harmonics of high-speed pinion frequency are the primary indicators of the defect.

BALANCING

86-1679
Dynamic Balancing with Micro Processors
D.G. Stadelbauer

Schenck Trebel Corporation, Deer Park, NY
Shock Vib. Dig., 18 (2), pp 3-6 (Feb 1986) 6 figs

KEY WORDS: Dynamic balancing, Microcomputers, Reviews

This article discusses the advantages of balancing using a micro processor. The balancing procedure is outlined and illustrated.

ANALYSIS AND DESIGN

ANALYTICAL METHODS

86-1680
Some Stability Results for General Linear Lumped-Parameter Dynamic Systems
M. Ahmadian, D.J. Inman
Clemson Univ., Clemson, SC
J. Appl. Mech., Trans. ASME, 53 (1), pp 10-14 (Mar 1986) 1 fig, 27 refs

KEY WORDS: Lumped parameter method, Stability, Lyapunov's method

A technique is presented for stability of equilibrium of general, linear, lumped-parameter dynamic systems. Liapunov functions are used to develop stability conditions that are direct in terms of the mass, damping, and stiffness matrices. The significance of what is presented here is twofold. First, this technique can be applied to general asymmetric systems. Second, it offers direct conditions that can easily be programmed on a digital computer to handle high-order systems. Many previously developed results, such as the KTC theorem and its extensions, are mentioned. Next, it is shown that the present study may provide broader applications because general systems are included and a more convenient approach is offered. Examples are used to illustrate the validity and applications of the presented results.

86-1681
Hierarchial Implicit Dynamic Least-Square Solution Algorithm
J. Padovan, J. Lackney
Univ. of Akron, Akron, OH
Computers Struc., 22 (3), pp 479-487 (1986) 6 figs, 2 tables, 16 refs

KEY WORDS: Finite element technique, Least squares method, Transient response

This paper develops an implicit type transient solution strategy which possesses hierarchial levels of application. In particular, due to the manner of formulation, stiffness updating, assembly inversion, solution constraint, as well as iteration are all performed at a localized level. The level of iterative calculations depends on the type of hierarchial partitioning employed, namely degree of freedom, nodal, elemental, material/nonlinear group, substructural, and so on. Since the iterative solution process and application of constraints are applied at a local level, the resulting so-called hierarchial implicit solution algorithm possesses very stable and efficient numerical properties and is highly storage efficient. To demonstrate the scheme, the results of several benchmark examples are presented. These enable comparisons with the Newton-Raphson solved implicit transient solution method. Overall the comparisons illustrate the superior stability and efficiency of the hierarchial scheme.

86-1682
Dynamic Analysis of Structures with Closely Spaced Modes Using the Response Spectrum Method
C. Manu
Control Data Canada, Ltd., Willowdale, Ontario, Canada
Computers Struc., 22 (3), pp 405-425 (1986) 13 figs, 12 tables, 17 refs

KEY WORDS: Response spectra, Modal superposition method, Geometric effects

An evaluation is made of some of the modal maxima superposition rules currently used to estimate the response of systems with closely spaced modes. Closely spaced modes arise in structures from geometrical effects -- such as symmetry and torsional unbalances -- and because of a light appendage with a frequency close to one of the natural frequencies of the structure. The proper use of the rules intended to account for closely spaced modes is reviewed. A slightly torsionally unbalanced structure is analyzed using response spectrum and time history methods. The results obtained show that only adequate modal maxima superposition can avoid too much conservatism. This does not mean that conservatism is bad, but only that in safety design, a rational evaluation of the margin of safety is the essential part of the work.

86-1683
Inclusion of Elastically Connected Members in Exact Buckling and Frequency Calculations
F.W. Williams, M.S. Anderson

UWIST, Cardiff, UK
Computers Struc., 22 (3), pp 395-397 (1986) 6 refs

KEY WORDS: Stiffness matrices, Eigenvalue problems, Natural frequencies

A standard stiffness matrix procedure which permits any combination of rigid, elastic, pinned or sliding connections of the degrees of freedom at the ends of a member to the nodes of its parent structure is described. It is extended to allow an existing algorithm to be used to ensure that no eigenvalues of the parent structure can be missed even when exact member theory is used. The eigenvalues are the natural frequencies of undamped free vibration analyses or the critical load factors of buckling problems. The method preserves the exactness of the member theory and an efficient method for computer application is indicated. The theory also permits any combination of rigid, elastic, pinned or sliding connections between the freedoms of a substructure and those of its parent structure.

86-1684
On the Monotonic Convergence of the Eigensolution by the h-Version of the Finite Element Method
D.A. Dunavant, C. Cheng
Purdue Univ., West Lafayette, IN
Computers Struc., 22 (3), pp 261-266 (1986) 8 figs, 8 refs

KEY WORDS: Finite element technique, Eigenvalue problems

The convergence of the strain energy of solutions by the finite element method (FEM) for static problems has been extensively investigated. It has been proven that for exactly and minimally conforming hierarchical element, convergence is monotonic as the number of degrees of freedom (NDOF) are increased by both the h-and p-versions. An investigation of the convergence of the eigensolution for dynamic problems by the h-version is presented. In general, an increase in the NDOF does not result in an improvement of the eigensolution. A condition for refinement is given under which monotonic convergence and improvement is ensured.

86-1685
Non-Normal Stochastic Response of Linear Systems
L.D. Lutes, Sau-Lon James Hu
Rice Univ., Houston, TX
ASCE J. Engrg. Mech., 112 (2), pp 127-141 (Feb 1986) 3 figs, 9 refs

KEY WORDS: Stochastic processes, Linear systems, Time domain method, Frequency domain method

Conventional analysis of mean squared values is extended to calculation of the fourth moment of the response of a linear system. This provides a way to investigate the non-normality of structural response, due to a non-normal excitation. Studies are made for a linear SDF structure subjected to fairly simple non-normal forces, which include a white noise process and filtered processes. Each filtered non-normal process is generated by passing a white noise non-normal process through a linear second-order filter. The non-normality of the structural response is investigated by integration in both the time domain and the frequency domain and by a more efficient technique that is an extension of the Lyapunov equation. This latter method reduces the calculation of the fourth moment of the response to solving a set of simultaneous linear algebraic equations.

86-1686
Mixed Time Integration for the Transient Analysis of Jointed Media
M.E. Plesha
Univ. of Wisconsin, Madison, Wisconsin
Intl. J. Numer. Anal. Methods Geomech., 10 (1), pp 91-110 (Jan/Feb 1986) 6 figs, 4 tables, 18 refs

KEY WORDS: Transient analysis, Rocks, Seismic analysis

A constitutive operator split method with implicit-explicit time integration is presented for the transient analysis of rigid block models of jointed media. The linear portion of the joint constitutive law is integrated by an implicit method and the non-linear, time dependent portion is integrated by an explicit method. the method features the stability of implicit procedures as well as the flexibility of explicit procedures for non-linear problems. The solutions obtained with this method are compared to the solutions obtained by the explicit central difference method; in all cases there is good to excellent agreement. For some problems, particularly for those with low frequency excitation, it is shown that the implicit-explicit method can result in a substantial savings over more conventional explicit methods.

86-1687
Theory and Application of Dynamic Decoupling in Structural Analysis
Toyoshiro Inamura
Kanazawa Univ., Kanazawa, Japan
Finite Elements Analysis Des., 1 (4), pp 323-339 (Dec 1985) 6 figs, 13 refs

KEY WORDS: Finite element technique, Structural modification techniques

A new method is proposed to decouple the dynamic characteristics of an undamped structure into those of the separated structures. This method is applicable to the results of dynamic analysis obtained by using the finite element method with lumped mass formulation. It is improved so that the decoupling can be done at high speed and on the lowest few modes of the structure. This method combined with the method of dynamic coupling proposed by the present author.

NUMERICAL METHODS

86-1688
A Time-Domain Method for Identifying Modal Parameters
Zhen-ni Wang, Tong Fang
Beijing Univ., Beijing, China
J. Appl. Mech., Trans. ASME, 53 (1), pp 28-32 (Mar 1986) 1 table, 13 refs

KEY WORDS: Modal analysis, Autoregressive/moving average model, Parameter identification technique, Time domain method

A time-domain method for identifying the modal parameters of a vibration system is presented. It is shown that system eigenvectors can be effectively estimated through the multivariate AR model representation of the system response to white noise excitation. In contrast to the usual ARMA model approach, in this method only a linear least square algorithm is required, so that a great amount of calculation is saved. Results of digital simulations support the identification method.

86-1689
Efficient Identification of Critical Stresses in Structures Subject to Dynamic Loads
R.V. Grandhi, R.T. Haftka, L.T. Watson
Virginia Polytechnic Institute and State Univ., Blacksburg, VA
Computers Struc., 22 (3), pp 373-386 (1986) 11 figs, 4 tables, 40 refs

KEY WORDS: Optimum design, Critical stress identification

Optimum structural design problems generally employ constraints which are parametric in terms of space and time variables. A parametric constraint may be replaced by equivalent critical point constraints at its local minima for optimization applications. In complex structures, accurate identification of such critical points is computationally expensive due to the cost of finite element analyses. Three techniques are described for efficiently and accurately identifying critical points for space- and time-dependent parametric constraints. An adaptive search technique and a spline interpolation technique are developed for exactly known response. A least squares spline approximation is suggested for noisy behavior. A helicopter tail-boom structure subjected to transient loading is used as an example to demonstrate the techniques described. All three techniques are shown to be computationally efficient for critical point identification and the least squares approximation also removes noise from the data. The case of multiple constraints per element is shown to be particularly suited to the use of spline techniques.

86-1690
Time Domain Parameter Identification Methods for Linear Modal Analysis: A Unifying Approach
J.M. Leuridan, D.L. Brown, R.J. Allemang
Univ. of Cincinnati, Cincinnati, OH
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 1-8 (Jan 1986) 8 figs, 2 tables, 33 refs

KEY WORDS: Parameter identification technique, Time domain method, Modal analysis

The paper describes a method that uses a multivariate model in the form of a nonhomogeneous finite difference equation to identify modal parameters of a mechanical structure. The modal parameters of this equation are estimable using a model that involves multiple input, multiple output vibration data. Thus, improved global estimates of modal parameters can be obtained, including the identification of highly coupled and pseudo-repeated modes of vibration. When the data are in the form of impulse or free decay response, then the parameters of the homogeneous part of the equation can be estimated separately, and the method is then related to the Least Square Complex Exponential method, the

Polyreference Time Domain method and the Ibrahim Time Domain method.

86-1691

Identification of Mass, Damping, and Stiffness Matrices of Mechanical Systems

C.-P. Fritzen
Univ. of Kaiserslautern, Fed. Rep. Germany
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 9-16 (Jan 1986) 10 figs, 4 tables, 19 refs

KEY WORDS: Matrix methods, Mass coefficients, Damping coefficients, Stiffness coefficients, Instrumental variable method

A procedure is presented to calculate the mass, damping, and stiffness matrices of mechanical systems from measured input/output data. It works on the basis of the Instrumental Variable Method which is well suited for the estimation of models from data with superimposed measurement noise. Noise is present in many practical cases. The theory of the method is described with regard to vibrating systems. The first application is the estimation of the matrices of a simulated system where the noise level is varied. The results show the expected properties: less sensitivity to noise compared to the Least Squares Method. Furthermore, the procedure is applied to a real system.

86-1692

Experimental Identification of Mechanical Structure with Characteristic Matrices

M. Ookuma, A. Nagamatsu
Tokyo Inst. of Tech., Tokyo, Japan
Bull. JSME, 28 (246), pp 2974-2977 (Dec 1985) 6 figs, 1 table

KEY WORDS: System identification techniques, Mass matrices, Damping coefficients, Stiffness matrices, Matrix methods

A new method of experimental identification is proposed in which a stable solution is always obtained for the real numerical calculation. Identification means the construction of a mathematical model which explains the dynamic characteristics of a machine or a mechanical structure. There are two ways in identification, namely theoretical and experimental ones. And there are two types of techniques for experimental identification. One is the type called curve fit which can identify the modal parameters. It has been studied by many researchers. The other type identifies characteristic matrices. But there are not many published reports about these techniques which are useful. In this report, a method of experimental identification of characteristic matrices is explained, and an example of numerical simulation is given.

86-1693

Three New Methods of Modal Identification

L. Jezequel
Ecole Centrale de Lyon, Ecully, France
J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 17-25 (Jan 1986) 9 figs, 4 tables, 23 refs

KEY WORDS: Parameter identification technique, Modal analysis

The identification of linear systems based on their dynamic responses has been the object of a great number of studies. This work proposes three new identification procedures which can be programmed on a microcomputer. The first of these methods uses an analytical extension of the transfer function, the second method uses a special integral transformation based on the Cauchy-Weierstrass theorem and the last method uses orthogonalization of the experimental displacement shapes by the Ritz-Galerkin procedure. These methods allow rapid detection of the modal parameters. Presented separately, they are applied to identical structure models defined in the first part of this study.

86-1694

An Efficient Method for Compensating Truncated Higher Modes in Structural Dynamics Modification

K.R. Chung, C.W. Lee
Korea Advanced Institute of Science and Technology, Seoul, Korea
IMechE. Proc., Part C: Mech. Engrg. Sci., 200 (C1), pp 41-48 (1986) 5 figs, 3 tables, 23 refs

KEY WORDS: Structural modification techniques, Truncation, Parameter identification technique

An efficient method for compensating the effects of the truncated higher modes in structural dynamics modification (SDM) is developed to predict the accurate modal parameters of locally modified structures. The effects of the truncated higher modes are represented by a fictitious, effective mode residing beyond the frequency range of interest. The modal parameters are then easily obtained by the iterative single

degree-of-freedom curve-fitting technique developed for lightly damped systems. A numerical example demonstrates the effectiveness of the improved SDM technique.

DESIGN TECHNIQUES

86-1695

A Constraint Function Technique for Improved Structural Dynamics

J.M. Starkey, J.E. Bernard
Purdue Univ., Lafayette, IN

J. Vib., Acoust., Stress, Rel. Des., Trans. ASME, 108 (1), pp 101-106 (Jan 1986) 8 figs, 12 refs

KEY WORDS: Structural modification techniques, Constraint function technique, Optimum design

A constraint function approach is presented for finding design changes that remove natural frequencies from undesirable frequency bands for lightly damped structures. The technique requires the minimization of a function which becomes smaller when natural frequencies clear out of undesirable bands, and design changes become small. Useful forms of these functions are defined, and the number of possible minima is explored. Graphical interpretations of the constraint functions are given, and an example is included which shows the effects of the parameters which weight these two functions.

AUTHOR INDEX

CALENDAR

SEPTEMBER

9-11 Rotating Machinery Dynamics, Bearings and Seals Symosium [Electric Power Research Institute, Coal Combustion Division] St. Louis, Missouri (Technical Information: Stanley Pace, Project Manager, CCS Division, Electric Power Research Institute, 3412 Hillview Ave., Palo Alto, CA 94304 415-855-2826; Registration information: Sharon Luongo, Conference Coordinator, Conferences & Travel Department, Electric Power Research Institute, 3412 Hillview Ave., Palo, Alto, CA 94304 415-855-2010)

14-17 International Conference on Rotordynamics [IFToMM and Japan Society of Mechanical Engineers] Tokyo, Japan (Japan Society of Mechanical Engineers, Sanshin Hokusei Bldg., 4-9, Yoyogi 2-chome, Shibuyak-ku, Tokyo, Japan)

16-18 Fall National Design Engineering Conference and Show (Cahners Exposition Group, New York, NY 203-964-0000)

21-23 Petroleum Workshop and Conference, Calgary, Canada (214-358-7601)

22-25 World Congress on Computational Mechanics [International Association of Computational Mechanics] Austin, Texas (WCCM/TICOM, The University of Texas at Austin, Austin, TX 78712)

29-30 VDI Vibrations Meeting [Society of German Engineers] Wurzburg, Fed. Rep. Germany (Society of German Engineers)

30-3 6th International Conference on Nondestructive Testing, Strasbourg, France (M.P. Pomes, 25 rue de Chong, 26500 Bourg les Valence, France)

OCTOBER

5-8 Mechanisms Conference [ASME] Columbus, OH (ASME)

7-9 2nd International Symposium on Shipboard Acoustics ISSA '86 [Institute of Applied Physics TNO] The Hague, The Netherlands (J. Buiten, Institute of Applied Physics TNO, P.O. Box 155, 2600 AD Delft, The Netherlands, Telephone: xx31 15787053, Telex: 38091 tpddt nl)

14-16 57th Shock and Vibration Symposium [Shock and Vibration Information Center] New Orleans, LA (Dr. J. Gordan Showalter, Acting Director, SVIC, Naval Research Lab., Code 5804, Washington, D.C. 20375-5000 - (202) 767-2220)

19-23 Power Generation Conference [ASME] Portland, OR (ASME)

20-22 Lubrication Conference [ASME] Pittsburgh, PA (ASME)

28-30 1986 41st Mechanical Failures Prevention Group Symposium, Patuxent River, MD (T. Robert Shives, A113 Materials Bldg., National Bureau of Standards, Gaithersburg, MD 20899)

NOVEMBER

3-6 14th Space Simulation Conference [IES, AIAA, ASTM, NASA] Baltimore, MD (Institute of Environmental Sciences, 940 E. Northwest Highway, Mt. Prospect, IL 60056 - (312) 255-1561)

7-14 Turbomachinery Symposium, Corpus Christi, TX (Turbomachinery Laboratories, Dept. of Mech. Engrg., Texas A & M Univ., College Station, TX 77843)

30-5 American Society of Mechanical Engineers, Winter Annual Meeting [ASME] San Francisco, CA (ASME)

DECEMBER

7-12 ASME Winter Annual Meeting, Anaheim, CA (ASME, United Engrg. Center, 345 East 45th Street, New York, NY 10017)

8-12 **ASA,** Anaheim, CA (Joie P. Jones, Dept. Radiology Sciences, Univ. of California, Irvine, CA 92717)

9-11 **ASA Fall Acoustical Show,** Anaheim, CA (katherine Cane, ASA Show Manager, Amer. Inst. of Physics, 335 E 45th St., New York, NY 10017)

1987

JANUARY

12-15 **AIAA 25th Aerospace Sciences Meeting,** Reno, NV

FEBRUARY

24-28 **SAE International Congress "Excellence in Engineering,"** Cobo Hall, Detroit, MI (SAE Engrg. Activities Div., 400 Commonwealth Drive, Warrendale, PA 15096)

APRIL

6-9 **5th International Modal Analysis Conference** [Union College and Imperial College of Science], London, England (IMAC, Union College, Graduate and Continuing Studies, Wells House -- 1 Union Ave., Schenectady, NY 12308)

6-8 **AIAA 28th Structures, Structural Dynamics and Materials Conference,** Monterey, CA

9-10 **AIAA Dynamics Specialist Conference,** Monterey, CA

JUNE

8-10 **AIAA 19th Fluid Dynamics, Plasma Dynamics and Laser Conference**

29-2 **AIIAA/SAE/ASME/ASEE 23rd Joint Propulsion Conference,** San Diego, CA

CALENDAR ACRONYM DEFINITIONS AND ADDRESSES OF SOCIETY HEADQUARTERS

AHS American Helicopter Society
1325 18 St. N.W.
Washington, D.C. 20036

AIAA American Institute of Aeronautics and Astronautics
1633 Broadway
New York, NY 10019

ASA Acoustical Society of America
335 E. 45th St.
New York, NY 10017

ASCE American Society of Civil Engineers
United Engineering Center
345 E. 47th St.
New York, NY 10017

ASLE American Society of Lubrication Engineers
838 Busse Highway
Park Ridge, IL 60068

ASME American Society of Mechanical Engineers
United Engineering Center
345 E. 47th St.
New York, NY 10017

ASTM American Society for Testing and Materials
1916 Race St.
Philadelphia, PA 19103

ICF International Congress on Fracture
Tohoku University
Sendai, Japan

IEEE Institute of Electrical and Electronics Engineers
United Engineering Center
345 E. 47th St.
New York, NY 10017

IES Institute of Environmental Sciences
940 E. Northwest Highway
Mt. Prospect, IL 60056

IMechE Institution of Mechanical Engineers
1 Birdcage Walk, Westminster
London SW1, UK

IFToMM International Federation for Theory of Machines and Mechanisms
U.S. Council for TMM
c/o Univ. Mass., Dept. ME
Amherst, MA 01002

INCE Institute of Noise Control Engineering
P.O. Box 3206, Arlington Branch
Poughkeepsie, NY 12603

ISA Instrument Society of America
67 Alexander Dr.
Research Triangle Pk., NC 27709

SAE Society of Automotive Engineers
400 Commonwealth Dr.
Warrendale, PA 15096

SEM Society for Experimental Mechanics (formerly Society for Experimental Stress Analysis)
7 School Street
Bethel, CT 06801

SEE Society of Environmental Engineers
Owles Hall
Buntingford, Hertz.
SG9 9PL, England

SNAME Society of Naval Architects and Marine Engineers
74 Trinity Pl.
New York, NY 10006

SPE Society of Petroleum Engineers
6200 N. Central Expressway
Dallas, TX 75206

SVIC Shock and Vibration Information Center
Naval Research Laboratory
Code 5804
Washington, D.C. 20375-5000

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Unsolicited articles are accepted for publication in the **Shock and Vibration Digest.** Feature articles should be tutorials and/or reviews of areas of interest to shock and vibration engineers. Literature review articles should provide a subjective critique/summary of papers, patents, proceedings, and reports of a pertinent topic in the shock and vibration field. A literature review should stress important recent technology. Only pertinent literature should be cited. Illustrations are encouraged. Detailed mathematical derivations are discouraged; rather, simple formulas representing results should be used. When complex formulas cannot be avoided, a functional form should be used so that readers will understand the interaction between parameters and variables.

Manuscripts must be typed (double-spaced) and figures attached. It is strongly recommended that line figures be rendered in ink or heavy pencil and neatly labeled. Photographs must be unscreened glossy black and white prints. The format for references shown in **Digest** articles is to be followed.

Manuscripts must begin with a brief abstract, or summary. Only material referred to in the text should be included in the list of References at the end of the article. References should be cited in text by consecutive numbers in brackets, as in the following example:

> Unfortunately, such information is often unreliable, particularly statistical data pertinent to a reliability assessment, as has been previously noted [1].
>
> Critical and certain related excitations were first applied to the problem of assessing system reliability almost a decade ago [2]. Since then, the variations that have been developed and practical applications that have been explored [3-7] indicate . . .

The format and style for the list of References at the end of the article are as follows:

- -- each citation number as it appears in text (not in alphabetical order)
- -- last name of author/editor followed by initials or first name
- -- titles of articles within quotations, titles of books underlined
- -- abbreviated title of journal in which article was published (see Periodicals Scanned list in January, June, and December issues)
- -- volume, issue number, and pages for journals; publisher for books
- -- year of publication in parentheses

A sample reference list is given below.

1. Platzer, M.F., "Transonic Blade Flutter -- A Survey," Shock Vib. Dig., 7 (7), pp 97-106 (July 1975).

2. Bisplinghoff, R.L., Ashley, H., and Halfman, R.L., Aeroelasticity, Addison-Wesley (1955).

3. Jones, W.P., (Ed.), "Manual on Aeroelasticity," Part II, Aerodynamic Aspects, Advisory Group Aeronaut. Res. Dev. (1962).

Articles for the **Digest** will be reviewed for technical content and edited for style and format. Before an article is submitted, the topic area should be cleared with the editors of the **Digest.** Literature review topics are assigned on a first come basis. Topics should be narrow and well-defined. Articles should be 3000 to 4000 words in length. For additional information on topics and editorial policies, please contact:

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